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DESIGN CRITERIA FOR DRY LUBRICATED FLIGHT CONTROL BEARINGS.(U)

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DESIGN CRITERIA FOR DRY LUBRICATED FLIGHT CONTROL BEARINGS

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May 1979

Final Report for Period June 1976 - March 1978

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Prepared for

APPLIED TECHNOLOGY LABORATORY
U. S. ARMY RESEARCH AND TECHNOLOGY LABORATORIES (AVRADCOM)
Fort Eustis, Va. 23604

APPLIED TECHNOLOGY LABORATORY POSITION STATEMENT

This report documents an attempt to develop wear-life equations for Teflon-lined rod end bearings used in helicopter flight control applications. The approach used in this program was unique, in that it was the first time that statistical design of experiments and statistical analysis of results were applied to the complex load/velocity regimes encountered in helicopter flight control applications. This approach, by reducing the number of required test specimens to a reasonable level, is economically viable and holds great promise for the development of an accurate wear-life predictive equation.

It is important to note that the equations that were developed during this program can be considered only as a first step toward a highly accurate predictor. They are not accurate enough to be used for wear calculations; however, they can be used as wear trend indicators. Additional testing will be required before these first-step equations can be evolved into highly accurate wear predictors. This laboratory is anticipating an in-house effort to accomplish this evolution.

The Technical Monitor for this program was Mr. Joseph D. Dickinson of the Applied Aeronautics Technical Area, Aeronautical Systems Division. Consultation regarding the statistical aspects of the work was provided by Mr. Timothy D. Evans of this laboratory.

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The objective of this program was to develop analytical methods for the selection of dry lubricated helicopter control bearings for specific conditions of use. The conditions of use include static radial pressure, cyclic radial pressure, speed of ball oscillation, angle of ball oscillation, static axial load, contamination, and combinations of these variables. Performance was measured by radial clearance which was directly related to wear.

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20. ABSTRACT (Continued)

Sixty-one bearings were tested in three wear test rigs. The wear test results were used to develop wear life equations. Three approaches to equation development were utilized in this report: empirical, deterministic, and theoretical.

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SUMMARY

The objective of the work performed under this contract was to develop analytical methods for the selection of dry-lubricated flight-control bearings for helicopter conditions of use. The conditions of use selected for investigation included appropriate ranges of static radial load, cyclic radial load, angle of ball oscillation, phase angle between cyclic radial load and angle of ball oscillation, speed of ball oscillation, static axial load, contamination (hostile environments) and combinations of these variables.

A statistically designed test program was performed to determine the effect on bearing-liner wear-life of these variables and their combinations. Wear-life equations were derived from the test results by statistical process. The wear-life equations were tested by selecting additional test conditions and running further tests.

It was found that the entire population of 48 test bearings consisted of two separate and distinct populations, namely: bearings with water contamination and bearings without water contamination. Within the specified limitations, the wear-life equations allow a designer to calculate wear and radial play with a statistically accepted confidence level. However, the designer is cautioned that the predictive equations may not be within ± 3 times the standard error of the estimate for the very short lives that result when water is present.

By rigorous definition, the wear-life equations are valid for the test conditions of this program: one size, one manufacturer and the variables tested over the ranges tested. However, test data are cited in Appendix E which indicate the equations will be valid over a range of sizes and for equivalent quality of manufacture, indicating that cautious general use of the wear-life equations may be made.

One portion of the program was to investigate the modification of an existing bearing wear-life equation to see if it could be made to be accurate for helicopter control system conditions. No successful modification was found. An equation was derived from the test data based on the well-known pressure x velocity (PV) approach. It was shown that PV is a significant variable. However, the PV equation explains only 32.8% and 63.2% of the variation in wear factor for the bearings tested without water and with water, respectively.

Another portion of the program investigated the incremental wear theory. This theory recognizes that a force is applied by the ball to the liner and the force travels a known distance. By definition, work is done. Some of the work results in wear. According to this approach, an increment of force which moves an incremental distance will produce an increment of wear. One corollary of this theory is that total wear is independent of the sequence with which various parts of a duty cycle are performed. Test data seems to confirm this corollary. This is an important concept because it allows the summing of wear lives from different load exposures during a mission profile.

A second corollary is that bearing-liner wear capacity can be measured by a very low-cost continuous-rotation wear test. This corollary was not proven or disproven because non-typical wear occurred during testing, a result of continuous-rotation test conditions that were inadvertently over-severe.

It was concluded that statistical design of test conditions, coupled with statistical analysis of test results, is a very useful procedure for investigating the many variables associated with helicopter control system bearing selection. The wear-life equations presented in this report represent a significant first step in improved helicopter flight-control bearing selection. It is recommended that the scope of the wear-life equations be increased by additional testing to evaluate the effects of bearing size, manufacturer, out-of-plane motion, water, and combinations of contaminants such as water with sand and dust and oil with sand and dust. Further, additional continuous rotation tests with lower energy input should be done to pursue this low-cost approach to bearing life testing.

PREFACE

The Applied Technology Laboratory at Fort Eustis, Virginia supervises research programs aimed at improving the life and reliability characteristics of components used on current and future Army helicopters. The prediction of wear life and reliability of dry lubricated flight control bearings used in helicopters has been identified as a problem area. Accordingly, the development of an improved analytical method of bearing selection for specific application requirements has been performed under Contract DAAJ02-76-C-0035. Technical direction was provided by Mr. Joseph McGarvey, Mr. M.B. Salomonsky, Mr. E.A. Birocco, and Mr. Joseph D. Dickinson, Aerospace Engineers at the Applied Technology Laboratory at Fort Eustis, Virginia. This report describes the test planning, the testing, the data analysis, and the development of the design equations. The test specimens were Type I Spherical Flight Control Bearings fabricated in conformance with MIL-B-81819, Draft #5 and MS14101-6 Bearing Standards by Rexnord Inc. Bearing Division, Downers Grove, Illinois. The program reported herein was accomplished during the period from 22 June 1976 to 28 February 1978. The work was conducted at Kaman Aerospace Corporation, Bloomfield, Connecticut under the technical supervision of Mr. Edward Nagy, Project Engineer. He was assisted in the design of the experiments and equation derivation by Mr. C.W. Carter, an industrial management consultant with considerable expertise in the statistical and factorially designed experimental testing field. Overall cognizance of the program was maintained by Mr. Robert B. Bossler, Jr., Chief of Mechanical Systems Research.

TABLE OF CONTENTS

TABLE OF CONTENTS (Continued)

TABLE OF CONTENTS (Continued)

TABLE OF CONTENTS (Continued)

LIST OF ILLUSTRATIONS

*NOTE: DL-XX(Y-Z) where XX is bearing serial number, Y is test run number, and Z is test bay number.

LIST OF ILLUSTRATIONS (Continued)

*NOTE: DL-XX(Y-Z) where XX is bearing serial number, Y is test run number, and Z is test bay number.

LIST OF ILLUSTRATIONS

LIST OF TABLES

LIST OF TABLES (Continued)

LIST OF TABLES (Continued)

INTRODUCTION

Rod end bearings are used extensively in helicopters, and a large percentage of these bearings are located in the flight control systems. The behavior of a helicopter in flight is peculiarly dependent on flight control bearings. Dry lubricated bearings are widely used because they are cost and weight effective. Characteristically, a ball is mounted within a surrounding metallic shell which has a spherical internal cavity fitted with a low friction wear surface usually made of some form of polytetrafluoroethylene (PTFE). This configuration permits relative motion about one, two or three axes; but, as with all bearings, internal clearance must be present in order to allow relative motion of the parts. Too much internal clearance, such as caused by wear, leads to lost motion between the pilot's input and the rotor(s) response. The lost motion often leads to pounding when loads reverse rapidly, usually with each revolution of the rotor, thus causing serious vibration problems. Bearing wear can accelerate in these conditions, making a bad situation worse and possibly leading to the eventual loss of the helicopter. Thus, considerable scrutiny is given to the selection and test evaluation of dry lubricated flight control bearings.

Unfortunately for the helicopter industry, the selection of dry lubricated flight control bearings is handicapped by a lack of information concerning their behavior under typical helicopter load, speed and environmental regimes. One can understand why detailed bearing performance data suitable for wear prediction is so sparse. The explanation is that there are very many factors which act and interact to produce wear in complex bearing applications. The relationships among these factors are imperfectly understood, so that their individual and interactive contributions to bearing wear have not been adequately described mathematically nor determined empirically.

The helicopter controls designer is painfully aware of the lack of good bearing design information for his applications, which involve relatively high sliding velocities and periodic vibratory or cyclic loading at different frequencies, in or out of phase with the sliding motion. These complex factors interact among themselves and with various environmental factors to make the prediction of bearing wear more an art than a science. Nonetheless, the control designer must come to grips with the problem. The techniques used vary with the individual, his experience, and the application. In general, however, the major considerations affecting the selection of a particular bearing include estimates of peak loads, oscillatory frequencies, the geometry of the particular application (will the bearing fit?) and a large measure of prior experience. The selection, then, is subject to change and modification as a result of expensive and time-consuming tests, and too frequently the changes are major.

The objective of the work performed under this contract was to develop analytical methods for the selection of dry lubricated helicopter flight control bearings for specific conditions of use. The conditions of use include static radial load, cyclic radial load, speed of ball oscillation, angle of ball oscillation, phase angle between cyclic radial load and angle of ball oscillation, static axial load, contamination and combinations of these variables. Performance was to be measured by radial clearance which is directly related to wear. Equations would be developed which would permit the designer to predict the wear expected from a particular application and therefore to select a bearing which has the reliability required.

TECHNIQUES FOR EQUATION DEVELOPMENT

We have used three general analytical techniques in developing a wear-life equation. These techniques are an empirical, a deterministic, and a theoretical approach to equation development. They are discussed separately below.

Preliminary to the task of developing a design equation, a series of test results must be obtained. These test results must be secured through a planned experiment in order to obtain answers in the most efficient and productive manner. The empirical approach was used to design most of our tests. The statistical methods of fraction-factorial experimentation which were utilized digressed from the primitive pattern of keeping all things constant while changing only one variable. Instead, the methods consisted of varying many factors within the constraints of statistical discipline. The technique evaluated the effect on the dependent variable, bearing wear, of each independent variable linearly and nonlinearly and the interactions of two or more independent variables. It identified and eliminated variables and combinations of variables which had negligible influence on bearing wear. The technique did not require massive amounts of testing. Because of the large number of independent variables, this approach was considered the most realistic. It gave better results with higher confidence at lower cost than any other method.

We have found no evidence of the application of statistical design of experiments or statistical analysis of results in the dry lubricated bearing industry. While it has been applied successfully in many other instances involving multiple variables, we seem to be plowing new ground with this particular application. Postulating the variables most important to wear and inferring their interactions requires engineering judgment and opinion. While we believe our understanding of these mechanisms is adequate for many purposes, the interactions described in the Taylor expansion equation and the test design allow impartial test results to determine the relative importance of the variables and their actions in the wear equation. This eliminates bias which would influence results in a conventional deterministic approach.

The second general analytical technique was the deterministic approach. This approach assumed that important variables and their interactions are known qualitatively. The work to be done was to express the equation quantitatively. Essentially, a presently used bearing selection equation was changed as required by the test data to produce a wear life equation for the particular type and make of bearing tested.

Finally, the third analytical technique was the theoretical approach. It is obvious that a force is applied by the ball to the liner and the force travels a known distance. By definition, work is done. Some of the work results in wear. We attempted to solve for the increment of wear produced by each increment of force as it moves through each increment of distance. This calculation is amenable to integral calculus and is easily adapted to computer calculation. Two important corollaries appear logical:

1. Total wear is independent of a series of required duties.

2. Wear life of a bearing can be assessed by a simple continuous rotation test wherein wear, pressure and distance-under-pressure are measured.

The first corollary is important because it allows the summing of wear lives from different load exposures during a mission profile. The second corollary is important because such testing is very fast, simple and cheap compared to conventional bearing evaluation testing.

EMPIRICAL APPROACH

INTRODUCTION

The empirical approach consisted of:

1. Identification of the primary operating parameters.
2. Identification of additional operating parameters.
3. Selection of variables to be tested.
4. Selection of variable levels.
5. Selection of statistical testing technique.
6. Development of experimental design for screening tests.
7. Completion of the screening tests (twelve-bay test rig).
8. Development of preliminary equations using stepwise multiple regression.
9. Selection of test conditions for validation tests.
10. Completion of the validation tests (twelve-bay test rig).
11. Development of final equations using stepwise multiple regression.

PRIMARY OPERATING PARAMETERS

The following factors can influence bearing wear, service life, and reliability:

Linear Sliding Velocity - This parameter is the velocity of the ball relative to the race (liner). This quantity is a function of the angle through which the ball rotates relative to the race, the frequency of this motion, and the ball's diameter.

Static Radial Load - Although control rod ends are operating in cyclic motion, they are usually subject to a static load during the entire cycle.

Cyclic Radial Load - This parameter varies with rotor azimuth position and has magnitude, frequency, and phasing relative to the sliding velocity.

Static Axial Load - This factor can be described as the force along the axis of the ball's bolt hole tending to dislodge the ball from the race. This factor can be significant because there is relatively little bearing surface to react the load.

Cyclic Axial Load - This load varies with rotor azimuth and has magnitude, frequency, and phasing relative to the sliding velocity.

Out-of-Plane Motion - This parameter can be described as the angle and phasing of motions out of the design plane of rotation, caused by the ball rotating about an axis other than the axis perpendicular to the plane of the race (wobble about the normal axis of rotation).

Temperature - MIL-B-81819 Draft #5 states by reference to MS14101 and MS14104 that the upper temperature limit is 325°F. However, temperatures below 325°F reduce the wear life of these bearings. Approaching wear-out, temperature increases at the liner surface, thus accelerating wear.

Contamination - The contaminants cited in MIL-B-81819 Draft #5 increase bearing wear depending on the make of bearing. In addition, P-D-680 Type I cleaning solvent, salt water, M2V hydraulic oil, and JP-5 fuel may increase bearing wear in certain liner systems.

ADDITIONAL OPERATING PARAMETERS

In addition to the previously mentioned parameters, the following parameters can be considered to affect bearing wear life in helicopter usage:

1. Cyclic radial load phase relative to the maximum linear velocity.
2. Cyclic axial load phase relative to the maximum linear velocity.
3. Cyclic radial load frequency relative to the ball oscillation frequency (1 per rev, 2 per rev, etc.).
4. Cyclic axial load frequency relative to ball oscillation frequency (1 per rev, 2 per rev, etc.).
5. Salt water contamination (This parameter can have especially severe effects on bearing performance, but Army helicopters normally do not operate in salt-laden air.)

6. Inner race or outer race rotation.

7. Bearing size.

8. Bearing manufacturer and liner system used.

9. Compression or tension loading.

10. Break-in.

11. Initial clearance (both radial and axial).

12. Interference fit.

13. Hours of usage.

14. Attitude (horizontal or vertical, shank up or down).

15. Bearing internal friction.

VARIABLES SELECTED FOR TESTING

Appendix A of this report describes the rationale used in the selection of the variables to be tested in this program. Of the various parameters previously listed, the following variables were selected for investigation:

1. Sand and dust contamination.

2. Cleaning solvent contamination.

3. Hydraulic fluid contamination.

4. Water contamination.

5. Static radial pressure.

6. Speed of ball oscillation.

7. Ball oscillating angle.

8. Cyclic radial pressure.

9. Static axial load.

10. Phase angle between the cyclic radial pressure and the ball oscillating angle.

11. Compression and tension loading.

12. Hours of usage.

SELECTION OF VARIABLE LEVELS

Once the variables had been identified, the number of levels and the values for these levels had to be determined for each variable. Table A-1 of Appendix A presents a list of the common helicopter ranges of variables and the four pages immediately following Table A-1 describe the reasoning used to select the levels. Contamination was listed as one variable with five conditions: none, sand and dust, cleaning solvent, hydraulic fluid, and water. Static radial pressure (psi) had four levels: 2000C, 1000C, 0, and 2000T. Speed of ball oscillation (cpm) had three levels: 300, 600, and 900. Ball oscillating angle (deg) had three levels: 5, 10, and 15. Cyclic radial pressure (psi) had four levels: 0, 1000, 1500, and 2000. Phase angle (deg) between ball oscillating angle and cyclic radial pressure had three levels: 0, 45, and 90. Static axial load (lb) had three levels: 0, 30, and 60. Finally, the hours of usage was set at 350 hours or failure, whichever came first.

STATISTICAL TESTING TECHNIQUE

To examine, in a full factorial experiment, the main effects and all interactions for the first seven variables and their respective levels as previously listed, a total of 5x4x3x3x4x3x3 (or 6480) tests would be required. The number of tests was reduced to a practical size by using a modified random assignment type of fraction factorial technique. Appendix A describes this technique and summarizes the major steps to develop the experimental design.

DEVELOPMENT OF EXPERIMENTAL DESIGN FOR SCREENING TESTS

Briefly, the fractional factorial technique that was used herein consisted of the following steps:

1. Select the number of tests to be performed in the screening test phase of the program. (In this program, 36 tests were selected as discussed in Appendix A).

2. Use a table of random numbers to help select the particular level to be used for each variable in each particular test. This is subject to the constraint of balance which calls for an equal number

of tests for each level of a given variable. The only exception was the variable contamination which had five conditions. It was decided that seven test specimens were to be tested with each of four contaminants and eight test specimens were to be tested with no contaminants.

3. Identify each of the 21 possible first-order interactions and any of the second-order interactions which have likelihood of being significant.

4. For each of the interactions identified, draw an interdesign interaction matrix with the number for each test located in the proper element of the matrix. See Figure 1 for interdesign BxF as an example. (Note that the ideal design consisted of an equal number of test specimens located in each element. At least 2 specimens should be provided, if possible, for replication purposes.)

5. Reassign test specimen numbers within each of the matrices wherein imbalance exists.

6. Finally, choose the test bays and the order of testing by the random number process, being careful to remain within the constraints of the test rig. Restrictions imposed by the test rig included:

   a. Only twelve bearings could be tested at one time.

   b. Of the twelve bearings in a above, four had to be tested at 300 cpm, four at 600 cpm, and four at 900 cpm.

The final conditions chosen for the screening tests are shown in Table 3 of this report and in Table A-3 of Appendix A.

BXF INTERACTION OF THE 36 TEST SPECIMENS

		F - STATIC AXIAL LOAD		
		F_1 = 0 LB	F_2 = 30 LB	F_3 = 60 LB
B- Speed of Ball Oscil- lation (CPM)	B_3=300	25, 30, 33, 36	28, 29	26, 27, 31, 32, 34, 35
	B_1=600	1, 3, 4, 8, 11, 12	2, 5, 6, 9	7, 10
	B_2=900	15, 22	13, 18, 20, 21, 23, 24	14, 16, 17, 19

Note: Numbers inside the matrix are the bearing test specimen numbers.

Figure 1. Interdesign interaction BXF (for 36 screening tests).

SELECTION OF TEST CONDITIONS FOR VALIDATION TESTS

In direct contrast to the random selection of conditions for the screening tests, the conditions for the validation tests on the twelve-bay test rig were selectively determined. Once the screening tests had been completed, a stepwise regression analysis, as described in Appendix B, was performed in order to obtain a preliminary equation for wear. This equation was used to calculate predicted wear values for all levels of the primary variables--contamination, static radial pressure, CPM, angle of oscillation, cyclic radial pressure, static axial load, phase angle--for 350 hours of test time. As discussed in Appendix C in greater detail, the total number of calculations was 10,125. From these predicted wear values, 78 conditions were chosen to represent both the highest and lowest calculated wear values. (Some of the lowest calculated wear values were even negative.) Eventually, twelve conditions were chosen for the validation tests. Please see Appendix C for a more detailed description of the selection process. The test conditions selected for the validation tests are shown in Table 5 and Table C-3 of Appendix C.

DETERMINISTIC APPROACH

INTRODUCTION

The deterministic approach utilizes existing bearing selection techniques modified as required by the test data. Suitable bearing selection equations were changed to produce a wear life equation for the particular type and make of bearing tested.

EQUATIONS SELECTED

Our literature search through the auspices of University of Connecticut's New England Research Application Center (NERAC) and other sources, although obviously not all-encompassing, unearthed various equations presented by bearing manufacturers. One of the best equations was given in nomograph form by Rexnord, Inc., Downers Grove, Illinois. The Rexnord equations were also chosen because they were presented for the Rexlon liner system, the very system possessed by the bearings tested in this program. The Equation Development section of this report describes the Rexnord equations and their subsequent modification.

As discussed in Appendix D of this report, our attempts to revise the Rexnord equations to fit our test data were not successful. After trying other equations, we finally had partial success with the frequently used PV-Wear factor equation. The modified equation produced predicted values which explained 32.8% and 63.2% of the variation in wear factor for the bearings tested without water and with water, respectively. The Equation Development section of this report describes the equation modification in detail.

THEORETICAL APPROACH

INTRODUCTION

Our theoretical approach to bearing wear life prediction relied upon a simple, basic energy concept. A force applied to the liner by the ball travels through a known distance and consequently work is done. Some part of this work results in wear. Within limits of load and velocity, wear is linear with distance.

The intent was to solve for the increment of wear produced by each increment of force as it moved through each increment of distance. Test data from the continuous rotation test rig was used for this calculation. The information needed to predict wear life was wear versus pressure at all anticipated velocities. According to the theory, with this information life can be calculated for any combination of static and dynamic loads acting at any relationship to any ball motion. The wear life is a function of the sum of the absolute values of the instantaneous pressure times the instantaneous velocity. This calculation is amenable to integral calculus and is easily adapted to computer calculation. Two important corollaries appear logical:

(1) Total wear life is independent of a series of required duties.

(2) Wear life of a bearing can be assessed by a simple continuous rotation test wherein wear, pressure and distance-under-pressure are measured.

The first corollary is important because it allows the summing of wear lives from different load exposures during a mission profile. The second corollary is important because such testing is very fast, simple and cheap compared to conventional bearing evaluation testing.

COROLLARY NO. 1

In an attempt to prove this corollary, tests of two identical MS14101-8 bearings, serial numbers 363 and 364, were conducted on the four-bay test rig from September 15, 1976 to November 12, 1976. Each bearing was tested a total of 638.5 hours and was subjected to approximately 128 hours of testing at each of five different load levels. The bearings were alternated from bay no. 1 to bay no. 2 so that each bearing had approximately 320 hours of testing in each of the two test bays. The test rig is described in the Test Program section of this report.

The radial load was always tension (never compression) and the five levels of load were 240, 480, 600, 720, and 960 pounds. The 600-pound load developed a static radial pressure of approximately 2000 psi. The oscillation frequency of the in-plane motion of the ball was 300 cpm with a ±10° angle of oscillation. No out-of-plane motion was present and no contamination was used. Zero cyclic radial pressure, zero static axial load, and zero phase angle were used.

COROLLARY NO. 2

In order to investigate this corollary, continuous rotation tests had to be performed.

An orthogonal Latin square design was chosen for the continuous rotation wear tests because a balanced design could be obtained with the four variables: static radial pressure, speed of ball rotation, static axial load, and out-of-plane motion. The initial pattern is shown in Table 1.

TABLE 1. LATIN SQUARE DESIGN

STATIC RADIAL PRESSURE	STATIC AXIAL LOAD	LOW SPEED		HIGH SPEED	
		OUT-OF-PLANE	IN-PLANE	OUT-OF-PLANE	IN-PLANE
HIGH	HIGH	X 49	X	X	X 55
	LOW	X	X 51	X 53	X
LOW	HIGH	X	X 52	X 54	X
	LOW	X 50	X	X	X 56

If one test specimen was provided for each element in the array, a total of 16 test specimens would be required. By providing only eight test specimens (Specimens #49 through #56), located as shown, the design became a half-replicate. The design still had balance, but the main effects have been confounded with the second-order interactions and the first-order interactions have been confounded with themselves.

SELECTION OF VARIABLE LEVELS

The Latin square design of Table 1 listed the four variables, each of which had two levels. The low static radial pressure was chosen to be 2000 psi compression because this was the radial pressure specified for the dynamic test requirements of MIL-B-81819, Draft #5 and was used by the Naval Air Development Center at Warminster, Pa. for the tests discussed in Appendix E of this report. The high static radial pressure was chosen to be 4000 psi compression because this was the maximum radial pressure ever applied to any of the test bearings in the twelve-bay test rig.

Next, the PV relationship was used to determine the high and low values for the speed of ball rotation. The maximum PV used in the screening tests in the twelve-bay test rig was 50,572 psi-fpm. A PV of this magnitude on the continuous rotation test rig would require a ball rotational speed of 77.3 rpm at the 4000 psi static radial pressure level. For simplification, the high value of ball rotational speed was chosen to be 75 rpm which produced a ball surface velocity of 12.27 fpm. The 75 rpm speed developed a PV of 49,086 and 24,543 psi-fpm for the 4000 psi and 2000 psi static radial pressure levels, respectively.

A PV of 32,725 psi-fpm was calculated for the -6 bearing dynamic test requirements of MIL-B-81819, Draft #5. A PV of this magnitude on the continuous rotation test rig would require a ball rotational speed of 50 rpm at the 4000 psi static radial pressure level. Thus, the low value of ball rotational speed was chosen to be 50 rpm, which produced a ball surface velocity of 8.18 fpm. The 50 rpm speed developed a PV of 32,725 and 16,362 psi-fpm for the 4000 psi and 2000 psi static radial pressure levels, respectively. It is important to note that the selection of the 50 and 75 rpm speeds produced four different PV test values that were in the ratio of 1, 1.5, 2, and 3 times the lowest value of 16,362.

The high static axial load was chosen to be 60 pounds, which was the highest static axial load used on the twelve-bay test rig. The low static axial load was chosen to be zero. The out-of-plane motion was chosen to be 6°, which was equivalent to the transient values listed for the transverse and rotational angles of oscillation in Table A-1, Appendix A, of this report.

The resultant list of test conditions is shown in Table 2. The Test Results section of this report describes the selection of test conditions for the validation tests.

TABLE 2. TEST CONDITIONS FOR CONTINUOUS ROTATION SCREENING TESTS

TEST NO.	BALL ROTATIONAL SPEED (rpm)	STATIC RADIAL PRESSURE (psi)	STATIC AXIAL LOAD (lb)	BALL ANGLE (deg)	BALL SURFACE VELOCITY (fpm)	PV (psi-fpm)
49	50	4000C	60	6	8.18	32,725
50	50	2000C	0	6	8.18	16,362
51	50	4000C	0	0	8.18	32,725
52	50	2000C	60	0	8.18	16,362
53	75	4000C	0	6	12.27	49,086
54	75	2000C	60	6	12.27	24,543
55	75	4000C	60	0	12.27	49,086
56	75	2000C	0	0	12.27	24,543

TEST PROGRAM

INTRODUCTION

In order to obtain the data for the equation development phase of this program, it was necessary to procure test bearings, install them in housings, modify test rigs, and conduct the tests.

OBJECTIVE

The overall objective of the test program was to measure the wear of the test bearings at various times as the bearings were being tested in order to be able to accurately determine the shape of the wear curve, including any break-in wear phenomenon. Specific test objectives were:

1. Obtain test results from 3 separate test runs using 12 test specimens per run for a total of 36 specimens.
2. Obtain test results from 4 separate runs using 2 specimens per run on the continuous rotation wear rig.
3. Obtain test results from 1 validation test run performed on each of 12 individual test specimens.
4. Obtain test results from 1 validation test run performed on each of 2 test specimens on the continuous rotation wear rig.
5. Obtain test results from 2 MS14101-8 bearings to be tested on the four-bay test rig.

DESCRIPTION OF TEST RIGS

The test specimens were tested in one of three wear test rigs. These three test rigs will be described in detail in the following subparagraphs.

A. Twelve-Bay Wear Test Rig - This test rig was a modification of the existing twelve-bay wear test rig which was used to test the wear-indicating rod end (WIRE) under USAAMRDL Contract DAAJ02-75-C-0031. (See Reference 1 for a complete description of the test rig with photographs included.) As shown in Figure 2, the wear test rig consisted of twelve

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1 Nagy, E.J., Wear-Indicating Rod End Bearing, USAAMRDL Technical Report 76-14, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia, September 1976, ADA030641.

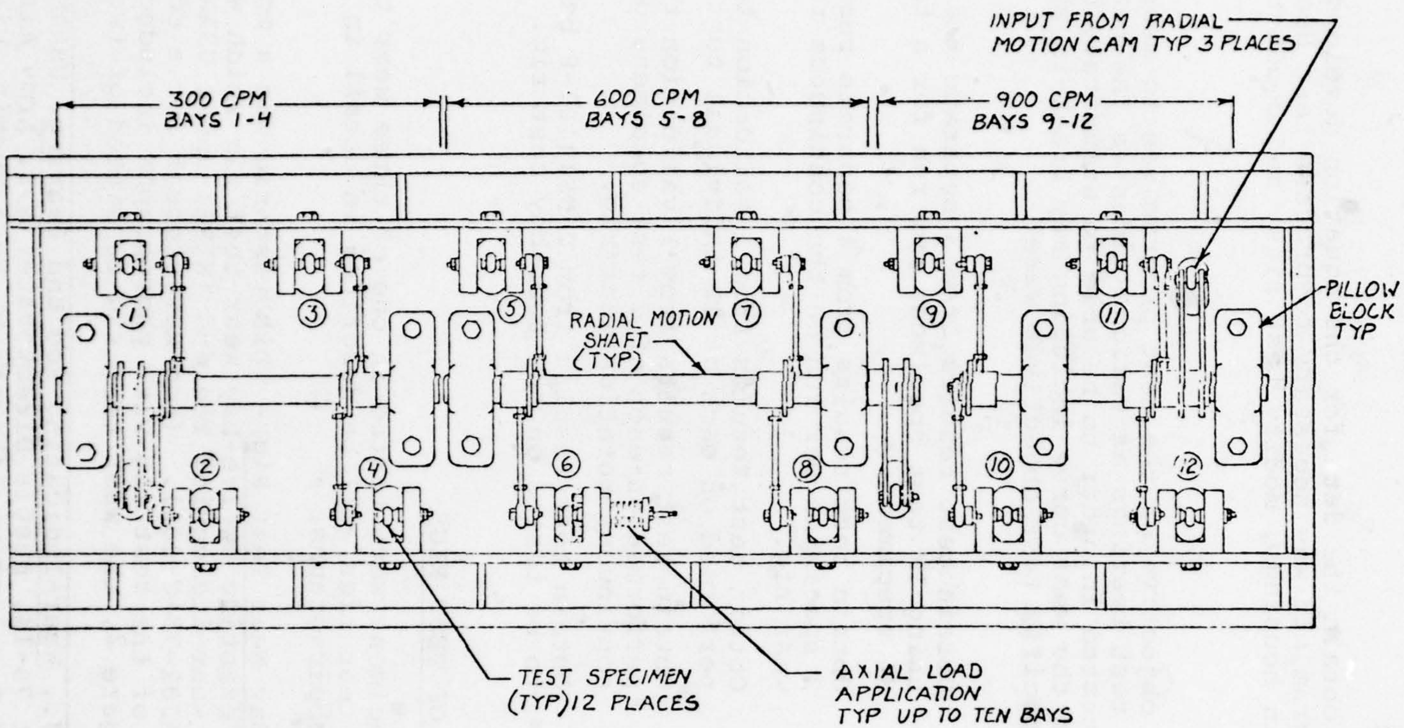

Figure 2. Twelve-bay test rig (plan view).

individual test stations. Each test station had provisions for testing one test specimen. The test specimen was positioned midway between two heavy-duty rig bearings. A suitable length NAS464-6 bolt was used to clamp two bushings against the ball of the bearing and to provide the radial load path.

As shown in Figure 3, radial load (both static and cyclic of either tension or compression) was provided by a shaft-driven cam coupled to a 2 to 1 load multiplier arm. The static load was adjusted with the turnbuckle and the cyclic load was adjusted with the cam.

For the bearings which were tested with zero cyclic radial load, the vibratory radial load cam was removed and replaced with a spring-bank assembly as shown in Figure 4. The spring was selected with a spring rate of approximately 150 pounds per inch. Thus, less time was required to check and readjust the loading for these test bearings.

The test specimen at each test station was thus radially loaded and held stationary while the ball was oscillated through the required angle by a torque arm which was also part of the aforementioned bolt-bushing-ball stack-up. The torque arms for bays #1, 2, 3, and 4 were driven by a common oscillating shaft and the radial load cams for bays #1, 2, 3, and 4 were driven at the same speed. The timing belts and the three 45° spaced holes on the vibratory radial load cam shaft allowed the phase angle between the cyclic radial load and the ball oscillating motion to be adjusted to 0°, 45°, or 90° for each test specimen. A similar setup was available for bays #5, 6, 7, and 8 as well as bays #9, 10, 11, and 12, except that bays #9, 10, 11, and 12 ran at a higher speed than bays #1, 2, 3, and 4. Bays #5, 6, 7, and 8 ran at a speed intermediate between the bays situated on both sides. The entire test rig was driven by a 7.5-hp vari-drive electric motor which allowed adjustment of the rig speed.

The ball angular oscillation was adjusted by positioning the horizontal push-rod in any of the three possible horizontal positions.

The axial load was applied to the test specimen using a setup as shown in Figure 5. The spring which was previously calibrated for compression load versus compressed length was used to apply the axial load to the ball of the bearing via a steel cable. The load was reacted by a steel spacer.

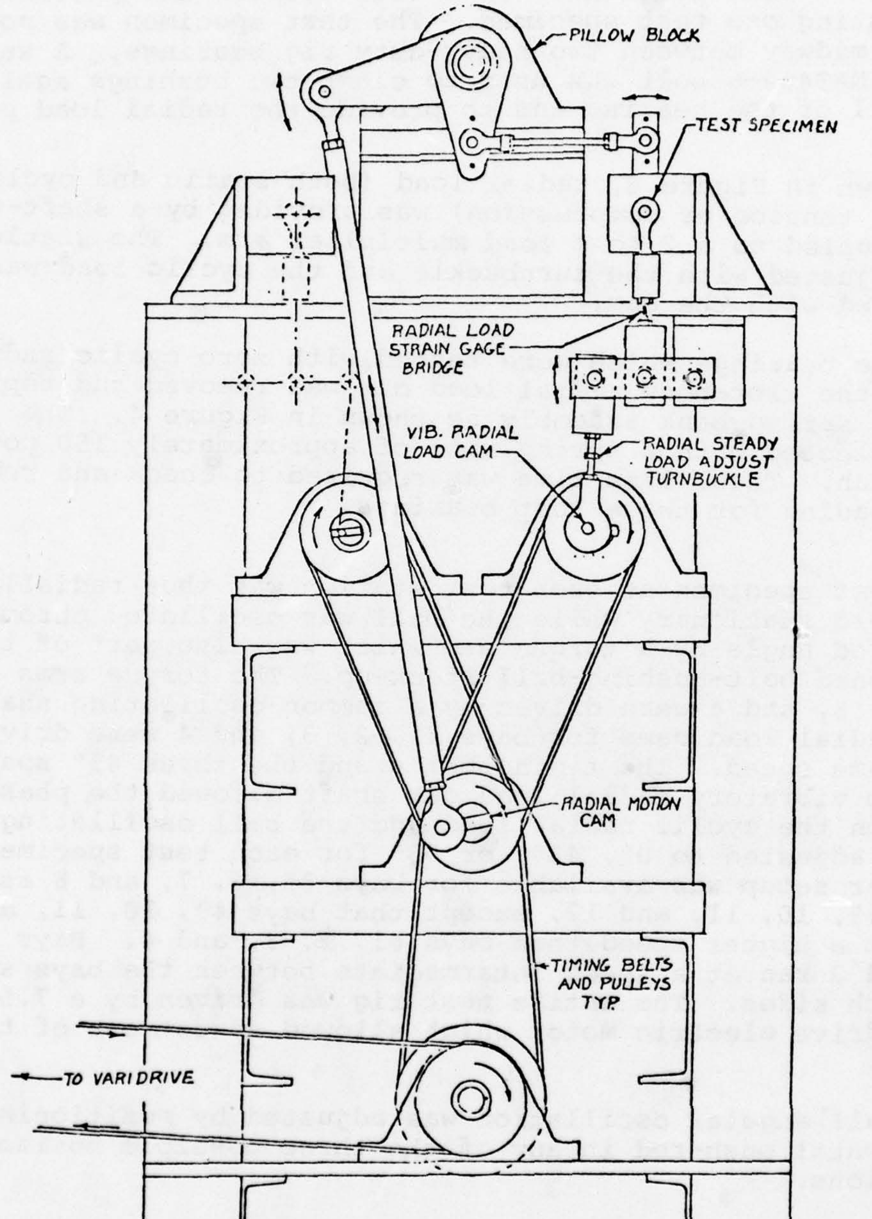

Figure 3. Twelve-bay test rig (elevation view).

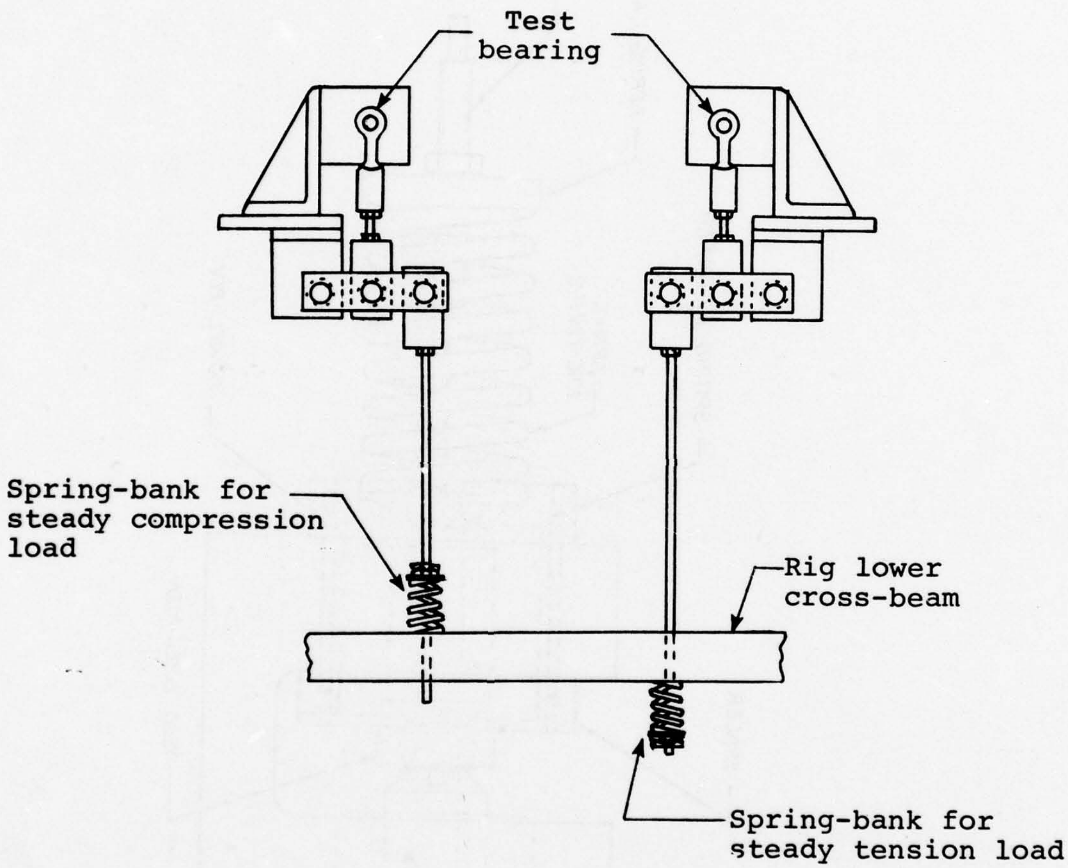

Figure 4. Spring-bank installation for zero cyclic radial load conditions.

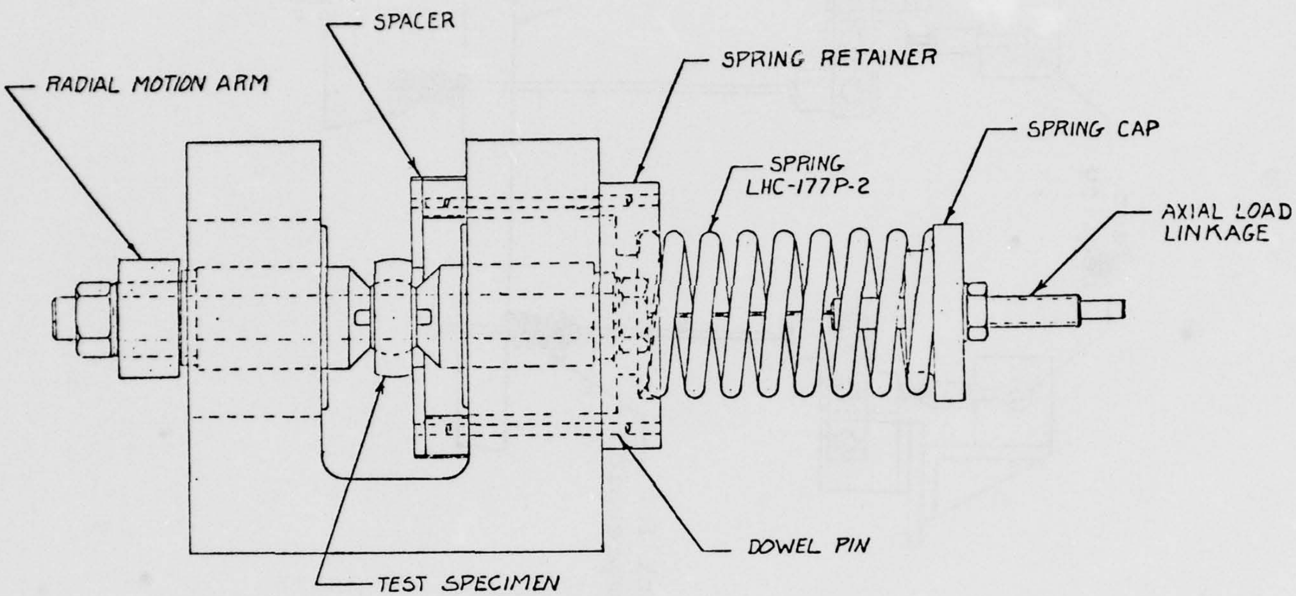

Figure 5. Axial load setup.

The water and sand and dust contaminants were manually applied to each side of the ball by either spraying or dusting while the ball was being oscillated. The contaminants were applied once each hour for 8 hours, followed by approximately 16 hours without contamination being applied, at which time the contamination cycle was repeated.

The load applied to each of the twelve test specimens was established and continuously verified by the use of twelve strain-gage bridges (one for each test specimen). These bridges were Poisson axial bridges located on the radial load application arm located directly below the specimen as shown in Figure 3. Each of the twelve bridges was calibrated against company standards. The signals from the individual bridges were fed through necessary signal-conditioning apparatus and read out on a CEC 12-inch oscillograph.

B. Continuous Rotation Wear Test Rig - As shown in Figure 6 the continuous rotation wear test rig had provisions for testing two test specimens simultaneously. Each test specimen was installed at the end of one of two continuously rotating shafts which were supported on suitable bearing pillow blocks. The shafts were driven through an adjustable sheave V-belt installation by a constant speed motor. Thus, both test bearings were tested at the same speed within the range of 50 rpm and 75 rpm. The two test stations each had provisions for application of steady axial load up to a maximum of 60 pounds with a cable-pulley deadweight system. The radial load at each test station was developed by deadweight applied through a force multiplying lever system and had radial load capability up to 642 pounds. Provisions were made available at each test station to align the ball for 0° (in-plane testing) or up to 6° (out-of-plane) testing.

C. Four-Bay Wear Test Rig - This rig is a four-bay constant radial load test rig. The location of the test specimen spaced midway between two rig bearings and the use of an NAS464 bolt for both clamping bushings against the ball of the test bearing and providing the radial load path make this test rig very similar to the twelve-bay rig. However, the test specimen is located such that the centerline of the bolt hole in the ball and the applied radial load lie in a horizontal plane rather than a vertical plane. The steady tension load was applied by a turnbuckle and monitored with a Chatillon dynamometer.

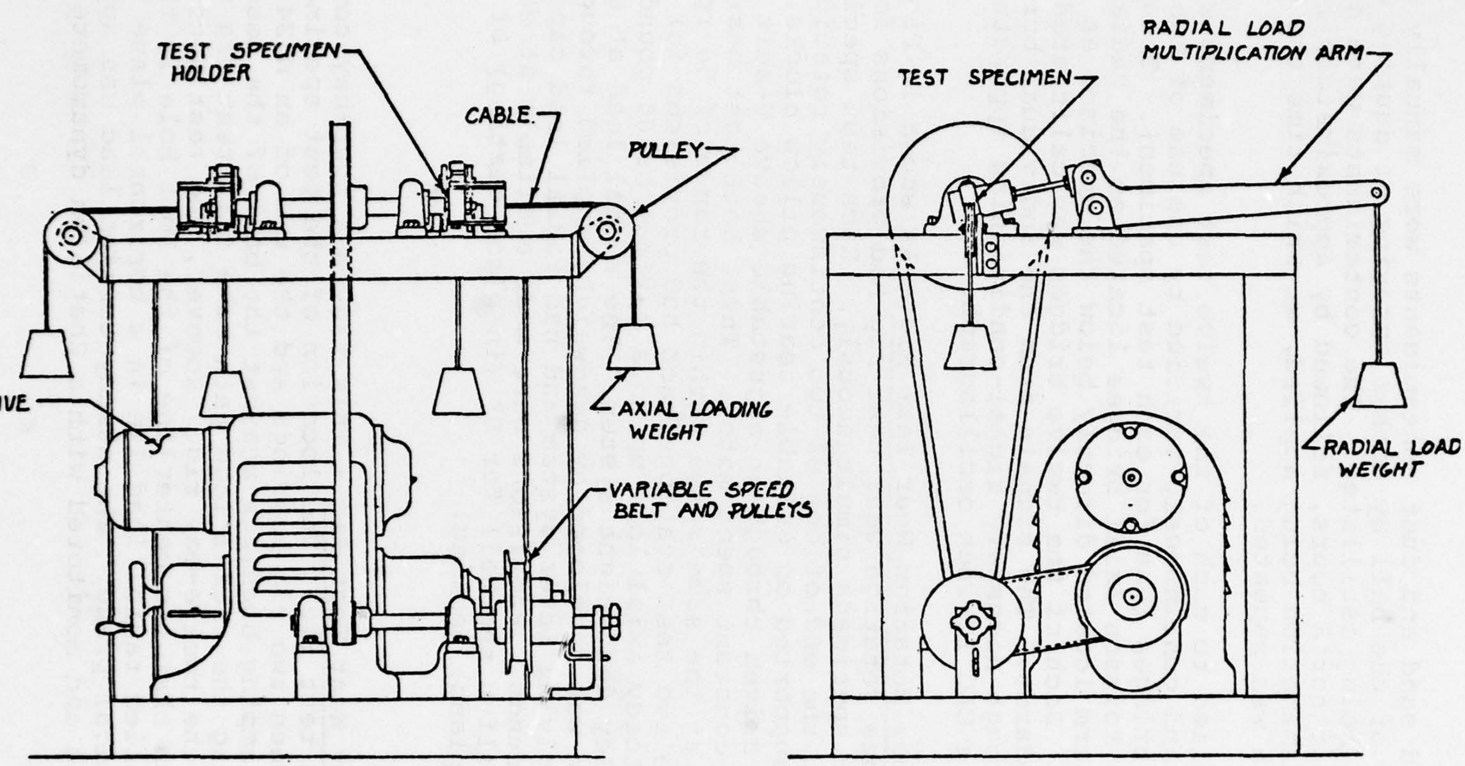

Figure 6. Continuous rotation wear test rig.

TEST SPECIMEN

The test bearings used in this program were limited by Request For Quote DAAJ02-76-Q-0004 to Type I spherical flight control bearings fabricated in general conformance with MIL-B-81819, Draft #5 and demonstrating wear performance consistent with bearings G-16-2 (Rexnord TGA-716, Rexlon -2) and G-16-8 (Kahr KNDB16-32, Kahrlon X12005) which were -16 size bearings tested by the Naval Air Development Center, Warminster, Pennsylvania to test condition #2 of Table II, page 7, MIL-B-81819, Draft #5. Since Request For Quote DAAJ02-76-Q-0004 also specified that existing test rigs be used, Kaman Aerospace Corp. proposed testing of -6 size bearings which could be easily tested in their existing twelve-bay test rig. Rexnord bearings were eventually used because the delivery schedule for these bearings was more consistent with the schedule of this program. Seventy-six bearings were obtained from Rexnord with part number TGA-706, Rexlon -2 liner system. These bearings were MS14101-6 bearings which had to be installed in Kaman manufactured rod ends for use in this program. Figure 7 depicts the rod end housing manufactured by Kaman Aerospace Corp. for the testing. A minimum of 14 housings were manufactured for use in this program. These housings were designed to be identical to the requirements of MIL-B-81819, Draft #5, Table II and Figure 3.

TEST PARAMETERS

The test specimens described above were tested in various conditions and in two different test rigs. Forty-nine test specimens were tested on the twelve-bay test rig and ten test specimens were tested in the continuous rotation test rig. The test conditions for these test specimens as well as for the two MS14101-8 bearings tested in the four-bay test rig follow.

A. Screening Tests in Twelve-Bay Test Rig - Table 3 lists the test conditions for the screening tests. (See Appendix A for a summary of the modified fraction factorial technique used to establish these test conditions.) As shown in Table 3, a total of 36 test specimens were tested in groups of 12. Column 2 of Table 3 specifies a sequence number of 1, 2, or 3 for each screening test, indicating whether the test specimen was tested in the first, second, or third group of 12. Column 3 indicates which bay of the test rig was used for each specimen. The remaining seven columns of Table 3 list the values of the seven variables to which the respective specimens were subjected. Table 4 lists the seven variables and their respective levels.

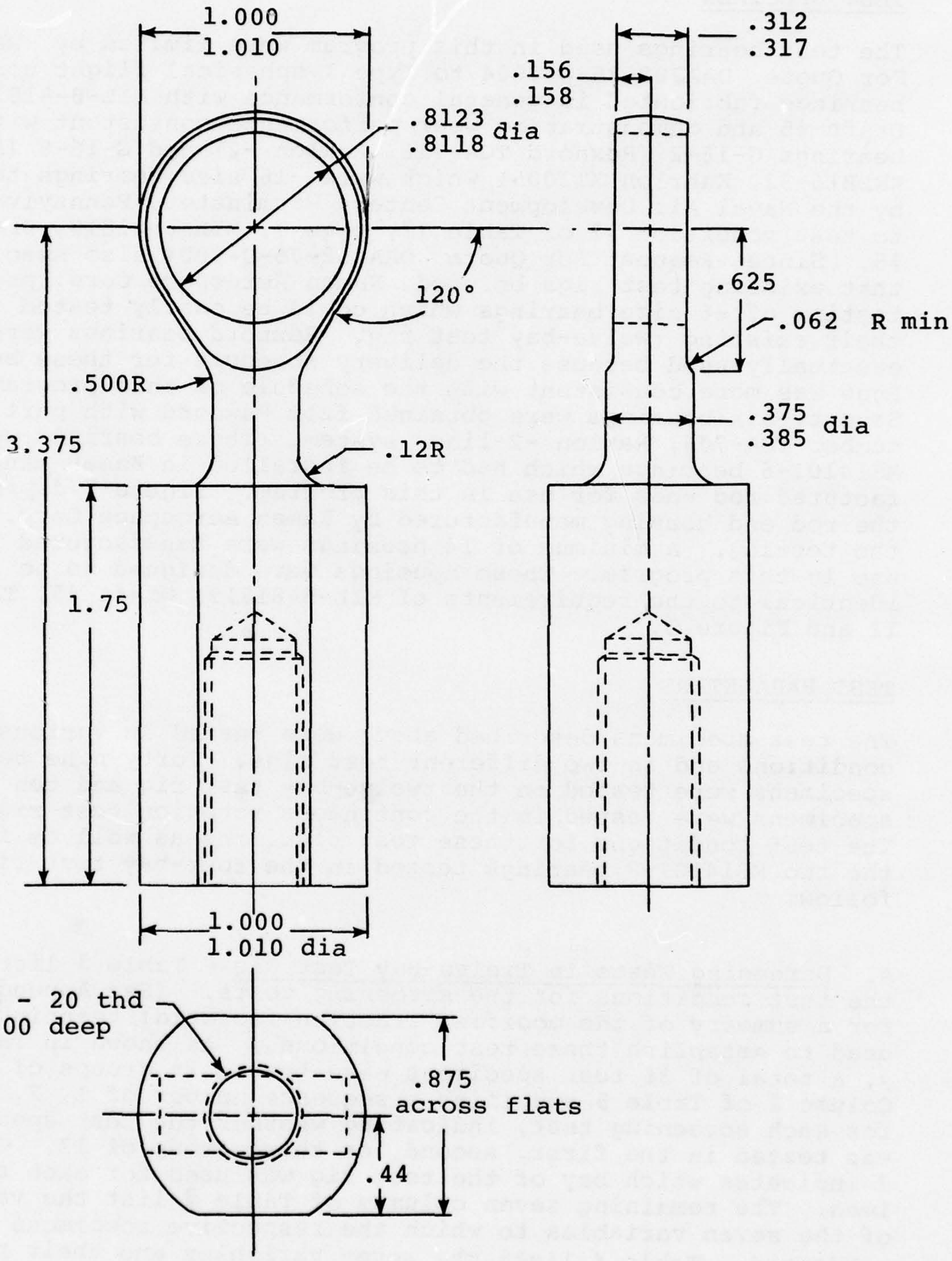

Figure 7. Bearing housing.

TABLE 3. TEST CONDITIONS FOR SCREENING TESTS IN TWELVE-BAY TEST RIG

			A	B	C	D	E	F	G
SCREENING TEST NO.	SEQUENCE NO.	BAY NO.	STATIC RADIAL LOAD (PSI)	CPM	ANG. OSC. (DEG)	CYCLIC RADIAL LOAD (PSI)	PHASE ANGLE (DEG)	STATIC AXIAL LOAD (LB)	CONTAM.
1		5	1000C	600	15	1000	0	0	680
2		6	2000C	"	10	0	0	30	None
3	1	7	2000T	"	5	1500	45	0	Water
4		8	2000T	"	10	1500	90	0	None
5		5	1000C	"	10	1000	45	30	S&D
6		6	0	"	10	1500	45	30	680
7	2	7	0	"	5	2000	45	60	None
8		8	0	"	15	2000	90	0	680
9		5	2000T	"	15	0	0	30	S&D
10		6	2000C	"	15	1500	90	60	Water
11	3	7	2000C	"	5	2000	45	0	500A
12		8	1000C	"	5	0	0	0	500A
13		9	0	900	10	1500	90	30	S&D
14		10	2000C	"	10	2000	90	60	Water
15	3	11	2000T	"	15	0	0	0	500A
16		12	2000C	"	5	1000	90	60	500A
17		9	1000C	"	15	1500	45	60	None
18		10	1000C	"	5	2000	90	30	S&D
19	1	11	2000T	"	5	0	0	60	S&D
20		12	2000C	"	15	0	0	30	680
21		9	1000C	"	10	0	0	30	680
22		10	2000T	"	10	1000	90	0	500A
23	2	11	0	"	5	1000	45	30	Water
24		12	0	"	15	1000	90	30	S&D

TABLE 3 (Continued). TEST CONDITIONS FOR SCREENING TESTS IN TWELVE-BAY TEST RIG

			A	B	C	D	E	F	G
SCREENING TEST NO.	SEQUENCE NO.	BAY NO.	STATIC RADIAL LOAD (PSI)	CPM	ANG. OSC. (DEG)	CYCLIC RADIAL LOAD (PSI)	PHASE ANGLE (DEG)	STATIC AXIAL LOAD (LB)	CONTAM.
25		1	2000C	300	10	1000	45	0	S&D
26		2	1000C	"	15	1000	45	60	Water
27	2	3	0	"	15	1500	0	60	680
28		4	0	"	10	2000	0	30	500A
29		1	2000T	"	15	2000	45	30	None
30		2	2000T	"	5	2000	45	0	680
31	3	3	1000C	"	5	2000	90	60	None
32		4	2000T	"	10	1000	90	60	None
33		1	0	"	5	1500	90	0	500A
34		2	1000C	"	10	0	0	60	Water
35	1	3	2000C	"	5	0	0	60	Water
36		4	2000C	"	15	1500	45	0	None

TABLE 4. LIST OF VARIABLES AND THEIR RESPECTIVE LEVELS

CODE	VARIABLE	LEVELS
A	Static Radial Pressure (psi)	2000C, 1000C, 0, 2000T (See Note 1)
B	Speed of Ball Oscillation (cpm)	300, 600, 900
C	Ball Oscillation Angle (deg)	±5, ±10, ±15
D	Cyclic Radial Pressure (psi)	0, ±1000, ±1500, ±2000
E	Phase Angle Between C & D (deg)	0, 45, 90
F	Static Axial Load (lb)	0, 30, 60
G	Contaminants	None, S&D, Water, 500A, 680 (See Notes 2 through 5)

NOTES:

1. "C" indicates that the radial load arm on the test rig will be in compression. "T" indicates tension.
2. S&D means sand and dust (Arizona Road Dust, P/N 1543094).
3. Water means distilled water.
4. 500A means Skydrol 500A hydraulic fluid.
5. 680 means P-D-680, Type I, dry cleaning solvent.

B. Validation Tests in Twelve-Bay Test Rig - Twelve test specimens were tested in the validation phase of the program. Table 5 lists the Test Conditions for the validation tests.

TABLE 5. TEST CONDITIONS FOR VALIDATION TESTS IN TWELVE-BAY TEST RIG

CASE NO.	VALIDATION TEST NO.	BAY NO.	STATIC RAD. PRESS.	CPM	ANG. OSC.	CYCLIC RAD. PRESS.	PHASE ANGLE	STATIC AXIAL LOAD	CONTAM.
*1	37	1	2000T	300	10	0	0	30	S&D
35	38	2	2000T	300	15	2000	90	60	500A
9a	39	3	2000C	300	5	0	0	60	500A
39	40	4	2000T	300	5	2000	90	60	680
4c	41	5	1000C	600	5	1000	0	0	Water
3a	42	6	1000T	600	5	2000	0	0	Water
26	43	7	2000C	600	10	1000	45	60	None
27	44	8	0	600	10	1500	45	0	680
2a	45	9	2000T	900	5	2000	0	0	Water
*2	46	10	0	900	10	1500	0	30	500A
1b	47	11	0	900	15	2000	0	0	Water
2d	48	12	2000C	900	15	2000	0	0	Water

C. <u>Screening Tests in Continuous Rotation Test Rig</u> - Eight bearings were tested at the test conditions shown in Table 6. (See the Theoretical Approach section of this report under the heading Corollary No. 2 for a summary of the Latin square design used to establish these test conditions.)

D. <u>Validation Tests in Continuous Rotation Test Rig</u> - The test conditions for the two validation tests are shown at the end of Table 6.

TABLE 6. TEST CONDITIONS FOR SCREENING AND VALIDATION TESTS IN CONTINUOUS ROTATION TEST RIG

	TEST NO.	BALL ROTATIONAL SPEED (cpm)	RADIAL LOAD (psi)	AXIAL LOAD (lb)	BALL ANGLE (deg)
Screening Tests	49	50	4000	60	6
	50	50	2000	0	6
	51	50	4000	0	0
	52	50	2000	60	0
	53	75	4000	0	6
	54	75	2000	60	6
	55	75	4000	60	0
	56	75	2000	0	0
Validation Tests	57	50	2000	60	6
	58	75	4000	0	0

E. Tests in Four-Bay Test Rig - Two MS14101-8 bearings, S/N 363 and 364, were tested in the four-bay test rig. As explained in detail in the Theoretical Approach section of this report under Corollary Number 1, the bearings were alternated from bay no. 1 to bay no. 2 and were tested at five different load levels. The five levels of load were 240, 480, 600, 720, and 960 pounds. The oscillation frequency was 300 cpm with a $\pm 10°$ angle of oscillation. No contamination was used. Zero cyclic radial pressure, zero static axial load, and zero phase angle were used.

TEST PROCEDURES

The procedures used were somewhat different for each test rig as discussed below.

A. Twelve-Bay Wear Test Rig - As described in the Test Parameters section of this report, four groups of twelve bearings each were tested in the twelve-bay wear test rig. (This included the thirty-six screening tests plus the twelve validation tests.) Before each group was tested the following was done:

1. Each of the twelve test specimens was installed in its respective test housing. (Figure 7 shows the housing configuration.)

2. Each of the twelve test specimens was installed in a radial wear measuring fixture for determination of initial radial wear tare readings. The radial wear measuring fixture is a device in which load reversal is applied to the ball of the test bearing in a radial direction and a dial indicator is used to measure radial displacement to 0.0001 inch.

3. Each of the twelve test specimens was installed in an axial play measuring fixture for determination of initial axial play readings. The axial play measuring fixture is a device in which load reversal is applied to the ball of the test bearing in the axial direction and a dial indicator is used to measure axial displacement to 0.0001 inch.

4. The test specimens which were to be contaminated with either Skydrol 500A hydraulic fluid or P-D-680, Type I dry cleaning solvent were immersed in their respective fluid, the test specimens were installed in their respective test bay, and no additional contamination was added for the duration of the test.

5. Each of the twelve test specimens was installed in its respective bay in the test rig and initial radial wear measurements were taken with a depth micrometer using a constant tension load and also a constant compression load. (The difference between the two readings was the initial radial play in the respective test specimen.) See the Data Reduction section of this report for greater detail on the depth micrometer readings.

6. The specific testing conditions (including static radial load, cyclic radial load, ball angular oscillation, phase angle, and static axial load) were adjusted for each individual test specimen.

7. The test rig was started and an oscillograph record was taken showing a time history of the radial loads being applied to each of the twelve test specimens. The test rig was stopped, the oscillograph record was analyzed, and any major discrepancies in the various test conditions were corrected.

Once the test conditions had been validated, the testing was begun and each of the twelve test specimens was tested for 350 hours or until failure, depending on which occurred first. Wear measurements were taken on each of the twelve test specimens after 6, 24, 50, 72, 100, 137, 175, 250, and 350 hours of test time.

After each test specimen had either failed or had reached the required 350 hours of testing, it was removed from the test rig and radial wear and axial play readings were obtained in the aforementioned wear measuring fixtures.

B. Continuous Rotation Wear Test Rig - Before the start of testing of any particular test specimen in this rig, the following were done:

1. Each bearing was installed in a steel test housing similar to that shown in Figure 7.

2. Each bearing was installed in the radial wear measuring fixture for determination of initial radial wear tare readings.

3. Each bearing was installed in the axial play measuring fixture for determination of the initial axial play.

4. Each test specimen was installed in its respective bay in the test rig.

5. The specific testing conditions were adjusted for the particular bearing.

Each bearing was tested for 200 hours or until failure, depending on which occurred first. At the approximate test times of 30, 60, 90, 120, 160, and 200 hours, the test bearings were removed from the test rig for wear measurement in the radial wear measuring fixture. In addition, axial play measurements were taken at the same measuring times for those bearings which had the 60-pound static axial load.

C. Four-Bay Wear Test Rig - The test bearings were installed in their respective housings and then installed in the test rig. Initial radial wear tare readings were taken with the radial test load applied. No axial play or radial play readings were taken on the wear measuring fixtures. Wear readings were taken without removal of the test bearings from the test rig and at the test times shown in Table 17.

DATA REDUCTION

A. Twelve-Bay Test Rig - Without removal of the test bearings from the test rig, wear measurements were taken at 0, 6, 24, 50, 72, 100, 137, 175, 250, and 350 hours of test time. At each measuring interval, the subject test bearing was placed in 200 pounds compression and depth micrometer readings were taken on a reference surface at the center of the bearing housing and also on the bolt on each side of the ball. As shown in Figure 8, the depth micrometer rested on the rig bearing pillow blocks which provided a good solid measuring reference surface. Depth micrometer readings D_L and D_R were taken by inserting the stem of the depth micrometer through a machined opening in each clamping bushing and making contact with the through-bolt on each side of the bearing ball. Use of readings D_L and D_R provided a means of eliminating bolt bending and rig bearing wear from the bearing wear measurement. The readings under compression load are identified as D_{LC}, D_{RC}, and D_{CC}. Subsequently, the subject test bearing was placed in 200 pounds tension and depth micrometer readings of D_{LT}, D_{RT}, and D_{CT} were obtained. Thus, the data consisted of six readings for each test bearing at each measuring interval.

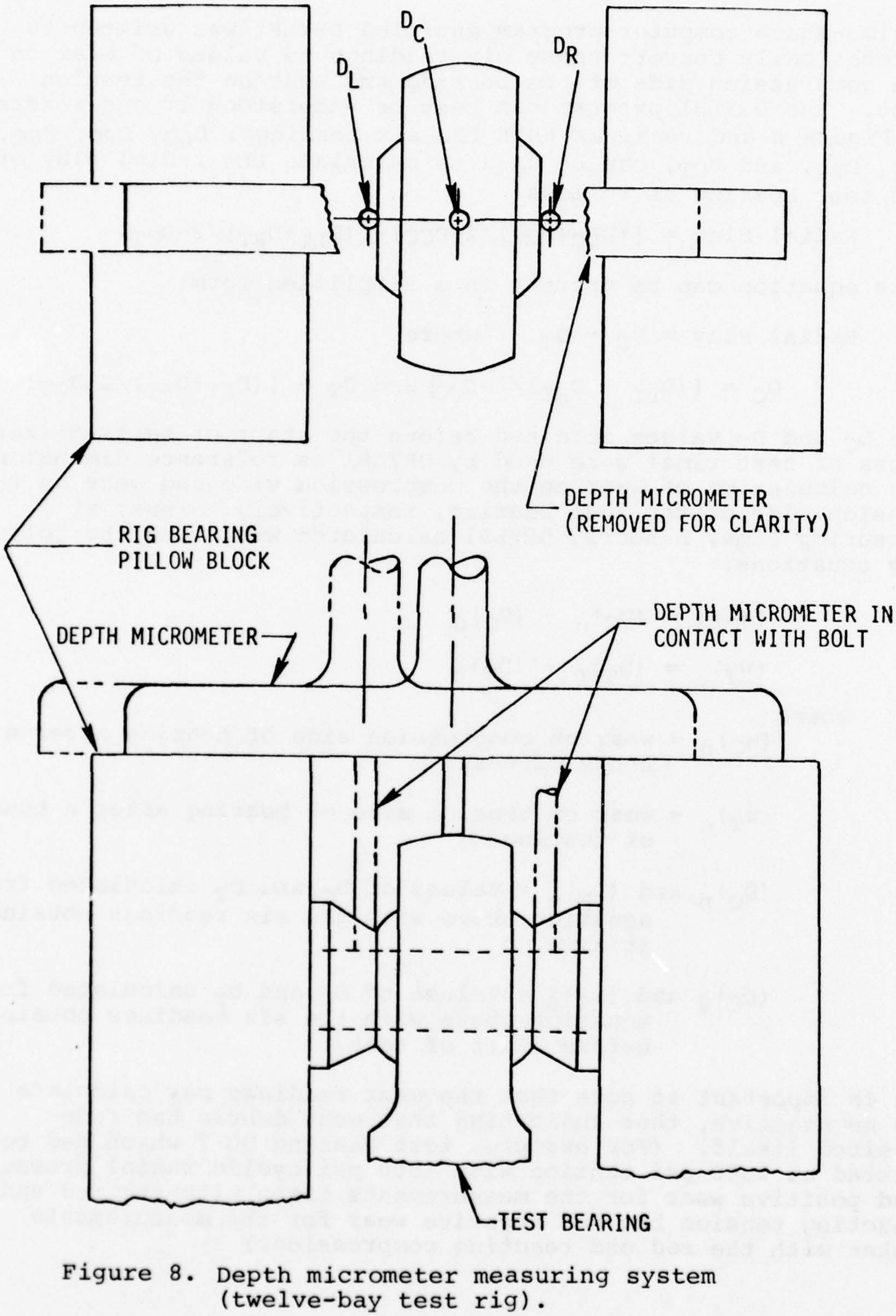

Figure 8. Depth micrometer measuring system (twelve-bay test rig).

A time-share computer program entitled DRYLB1 was written to automatically convert these six readings to values of wear on the compression side of the bearing and wear on the tension side. The DRYLB1 program can best be understood if one refers to Figure 8 and realizes that the six readings, D_{LC}, D_{RC}, D_{CC}, D_{LT}, D_{RT}, and D_{CT}, can be used to calculate the radial play of the test bearing as follows:

$$\text{Radial Play} = [(D_{LC}+D_{RC})/2-D_{CC}]-[(D_{LT}+D_{RT})/2-D_{CT}]$$

This equation can be written in a simplified form:

$$\text{Radial Play} = D_C - D_T \quad \text{where}$$

$$D_C = [(D_{LC} + D_{RC})/2-D_{CC}] \text{ and } D_T = [(D_{LT}+D_{RT})/2-D_{CT}]$$

The D_C and D_T values obtained before the start of testing (zero hours of test time) were used by DRYLB1 as reference dimensions for calculation of wear on the compression side and wear on the tension side of the test bearing, respectively. Thus, at measuring time, n hours, DRYLB1 calculated wear with the following equations:

$$(W_C)_n = (D_C)_n - (D_C)_0$$

$$(W_T)_n = (D_T)_0 - (D_T)_n$$

where

$(W_C)_n$ = wear on compression side of bearing after n hours of testing

$(W_T)_n$ = wear on tension side of bearing after n hours of testing

$(D_C)_n$ and $(D_T)_n$ = values of D_C and D_T calculated from equation above with the six readings obtained at time, n

$(D_C)_0$ and $(D_T)_0$ = values of D_C and D_T calculated from equation above with the six readings obtained before start of test.

It is important to note that the wear readings may calculate to be negative, thus indicating that wear debris has redeposited itself. (For example, test bearing DL-7 which had been tested at 2000 psi tension with 1500 psi cyclic radial pressure had positive wear for the measurements taken with the rod end reacting tension but had negative wear for the measurements taken with the rod end reacting compression.)

B. Four-Bay Test Rig - Only readings D_{LT}, D_{RT}, and D_{CT} were taken. No readings were taken with the bearing in compression. The equation, $(W_T)_n = (D_T)_0 - (D_T)_n$, which was described for the twelve-bay test rig was used to obtain test bearing wear.

TEST RESULTS

Sixty-one bearings were tested in this program, of which fifty-nine were Rexnord bearings. The bearings performed exceptionally well in spite of the frictional heating encountered in the first couple of hours of testing on the twelve-bay test rig. The Rexnord bearings had minimal clearance and could be classified as "tight" bearings. (See Appendix F for a tabulation of initial radial and axial play readings for 48 of the Rexnord bearings.) It is postulated that the minimal clearance condition caused these bearings to run hot enough to burn a finger and the bearings did not cool down until additional clearance had been produced by the wear-in process. (In contrast, the bearings tested in the continuous rotation test rig ran hot throughout their entire test runs.) No deleterious effects were noted because of the frictional heating encountered on the twelve-bay test rig. This cannot be said for the continuous rotation test rig and will be discussed in detail later.

Forty-nine bearings were tested in the twelve-bay test rig and thirty-eight of them survived the planned 350-hour test run. One additional bearing, DL-14 (1-2)[a], had also survived the planned test run but was classified as a failure because of the high radial wear and axial play readings exhibited after test completion. Thirteen of the forty-nine bearings were tested with water. Not only did these bearings experience the highest wear values, but also they had essentially failed by the end of the 350-hour test run. Six of the thirteen water contaminated bearings had a static axial load of 60 pounds applied (DL-27(2-11)[a] had only 30 pounds), and these bearings had high axial wear. In contrast, bearing DL-7(1-7A)[a] had no static axial load, low axial wear, but high radial wear. It is important to note that high wear caused by water contamination is common to all Teflon bearings and therefore is not peculiar to this particular type of bearing.

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a Note: The bearings will be designated by the following format:

DL-XX(Y-Z) where

DL-XX = bearing serial number
Y = run number (4 runs performed)
Z = bay number in which the bearing was tested

WEAR MEASUREMENTS (TWELVE-BAY TEST RIG)

Wear measurements were the main outcome obtained from the tests in the twelve-bay test rig. Table 7 presents the measured wear (corrected for rig bearing wear) for the portion of the bearing liner which is loaded when the rod end shank is in compression. Table 8 presents the measured wear (corrected for rig bearing wear) for the portion of the bearing liner which is loaded when the rod end shank is in tension. It can be noted that negative wear values show up in both tables. This is the direct result of wear debris being redeposited into a region of the bearing wherein there is either no load or a reduced level of load. The effect is most pronounced in a bearing which is loaded in steady compression (or tension). The negative wear will appear on the unloaded side of the bearing. Because of this phenomenon, radial play and total bearing wear are not always equal. Radial play for any particular bearing is the summation of the wear value listed in Table 7 and the corresponding wear value in Table 8. Take, for example, test bearing 4-1 which was tested at 2000 psi tension with zero cyclic radial pressure. At 100 hours the bearing had experienced .00035 inch of wear on the loaded side (tension), but had also undergone .00020 inch of negative wear due to redeposition of wear debris to the unloaded side (compression). The radial play at 100 hours was .00015 inch, but the total wear was .00035 inch. (Please see note c on Table 9.)

Total bearing wear may be a difficult concept to grasp. For bearings loaded primarily in one direction wherein the load may reduce to zero but does not reverse direction, total bearing wear equals the wear measured on the loaded side. For bearings with reversing loads, total bearing wear is the summation of the wear measured in both directions (for instance, test bearing 2-11 which had zero static radial pressure and 1500 psi cyclic radial pressure). At 100 hours the wear on the compression side was .00120 inch and the wear on the tension side was .00070 inch. Therefore, total bearing wear is the summation of those two values, or .00190 inch. Radial play also is .00190 inch for this particular bearing. Table 9 presents the total bearing wear for all 49 test bearings.

PLOTS OF WEAR VERSUS TIME (TWELVE-BAY TEST RIG)

Figures 9 through 21 are plots of liner wear versus time for the thirteen bearings tested with water contamination, plus bearing DL-40(3-5) tested with sand and dust. Each plot presents the wear on the compression side, the wear on the tension side, and total bearing wear. The plots for the remaining 35 bearings were not presented because of the low values of wear encountered.

TABLE 7. WEAR READINGS ON COMPRESSION SIDE

(INCHES x 10^5)

RUN NO.	BAY NO.	Time in Hours: 6	24	50	72	100	137	175	250	361
	1	25	5	20	35	15	5	0	20	40
	2	5	-5	10	55	225	355	470	805	2955
1	3	10	5	0	130	290	570	855	1760	3280 (325)a
	4	-	45	30	45	45	50	55	50	60
	5	-	-5	35	85	85	75	110	135	165
	6	50	60	60	60	65	105	110	135	155
	7A*	-	5	-50	-85	-230	-295	-560	-1125 (189)a	-
	8	-	-40	-45	-55	-40	-55	-50	-75	-40
1	9	-	60	75	85	85	120	135	140	225
	10	-	75	45	45	80	85	80	75	70
	11	-	-	-	-	-	-	-	-	-
	12	30	5	20	65	70	75	115	140	155

RUN NO.	BAY NO.	Time in Hours: 11	24	47	72	102	137	158
1	7B*	-15	-5	-85	-125	-220	-410	-460

RUN NO.	BAY NO.	Time in Hours: 6	24	56	72	100	136	175	250	350
	1	60	75	70	75	85	90	85	65	65
	2	20	15	50	215	390	515	830	1105 (244)a	-
2	3	-40	-15	25	0	10	-30	-45	25	15
	4	-15	20	15	25	10	80	85	90	90
	5	60	55	45	50	10	35	60	95	85
	6	-50	-50	-50	-30	-50	-50	-30	-55	-40

*Early failure in bay 7 allowed a second run. The two runs are identified as 7A and 7B.

TABLE 7 (Continued). WEAR READINGS ON COMPRESSION SIDE

(INCHES x 10^5)

RUN NO.	BAY NO.	Time in Hours 6	24	56	72	100	136	175	250	350
2	7	30	10	15	0	0	5	0	5	10
	8	45	15	40	40	10	100	95	70	85
	9	0	30	15	0	30	5	65	65	120
	10	-80	-50	-15	-65	-90	-90	-110	-90	-105
	11	-5	0	90	115	120	85	180	200	135 (267)[a]
	12	0	0	25	30	25	20	-10	30	50

RUN NO.	BAY NO.	Time in Hours 6	24	50	72	100	137	175	250	350
3	1	-10	-20	-15	0	0	0	-10	-10	-10
	2	-5	-5	-10	-10	-5	-15	-15	-15	0
	3	15	20	25	15	15	10	15	25	35
	4	20	10	10	-20	10	0	0	5	5
	5	-	-	-	-	-	-	-	-	-
	6	55	110	225	315	630	940	1310	2745 (191)[a]	-
3	7	60	80	75	75	90	105	105	110	115
	8	60	60	85	100	95	100	95	135	160
	9	-95	-65	-65	-65	-30	25	25	40	80
	10	50	70	90	140	175	305	565	1880 (214)[a]	-
	11	-	-	-	-	-	-	-	-	-
	12	55	65	80	90	90	100	100	105	145

TABLE 7 (Continued). WEAR READINGS ON COMPRESSION SIDE

(INCHES x 10^5)

RUN NO.	BAY NO.	6	24	50	72	100	137	175	250	350
					Time in Hours					
	1	-10	-15	-5	-15	-20	-30	-40	-65	-110
	2	-55	-70	-70	-75	-75	-80	-100	-110	-120
4	3	15	45	60	70	75	65	80	90	105
	4	5	10	5	10	5	10	-5	0	5
	5	5	30	215	400	450	475	725	1115	1880
	6	-100	-20	45	70	140	210	270	300	365
	7	60	70	80	65	70	100	75	115	130
	8	65	80	85	125	95	95	95	115	115
4	9	-65	-70	-195	-280	-390	-755	-1000 (168)a	-	-
	10	95	85	75	85	80	80	70	-55	-35
	11	10	130	230	330	450	655	830	900 (193)a	-
	12	95	170	275	440	665	875	990	1355	3115 (271)a

a Note: Testing of bearings 1-3, 1-7A, 2-2, 2-11, 3-6, 3-10, 4-9, 4-11, and 4-12 was discontinued after the test hours noted within the respective parentheses.

TABLE 8. WEAR READINGS ON TENSION SIDE

(INCHES x 10^5)

RUN NO.	BAY NO.	Time in Hours								
		6	24	50	72	100	137	175	250	361
	1	-	-5	5	-5	10	0	-5	15	-25
	2	-	-	-	-	-	-	-	-	-
	3	-	-	-	-	-	-	-	-	-
1	4	-	-25	-55	-25	-25	-5	0	-25	-45
	5	20	45	20	15	15	25	5	20	5
	6	-	-	-	-	-	-	-	-	-
	7A*	-	50	145	460	835	1175	1645	2400 (189)[a]	-
	8	-	45	55	70	70	105	95	115	125
	9	-	15	-10	-10	10	20	20	15	10
1	10	5	-40	-65	-60	-55	-25	-10	-25	-20
	11	20	15	-5	15	25	35	35	155	230
	12	-	-	-	-	-	-	-	-	-
		Time in Hours								
		11	24	47	72	102	137	158		
1	7B*	70	165	360	555	775	1090	1165		
		Time in Hours								
		6	24	56	72	100	136	175	250	350
	1	-65	-75	-80	-80	-70	-75	-85	-70	-85
	2	-55	-45	-40	-70	-30	-120	-245	-615 (244)[a]	
2	3	15	10	55	15	-15	10	0	-30	-35
	4	45	40	65	90	85	0	-45	-5	10
	5	-50	-35	-25	-40	-35	-50	-50	-10	-10
	6	55	70	75	70	55	25	25	70	60

*Early failure in bay 7 allowed a second run. The two runs are identified as 7A and 7B.

TABLE 8 (Continued). WEAR READINGS ON TENSION SIDE

(INCHES x 10^5)

RUN NO.	BAY NO.	Time in Hours: 6	24	56	72	100	136	175	250	350
	7	-15	10	20	20	20	20	10	20	15
	8	0	15	10	-30	-5	-35	-65	-45	-50
2	9	-	-	-	-	-	-	-	-	-
	10	60	65	75	70	95	115	105	140	140
	11	25	10	95	75	70	70	115	180	125 (267)a
	12	25	15	15	45	40	20	-15	25	45

RUN NO.	BAY NO.	Time in Hours: 6	24	50	72	100	137	175	250	350
	1	15	10	20	10	25	30	25	5	35
	2	20	35	35	45	35	35	35	20	10
3	3	-15	-10	-20	-25	-15	-20	-25	-55	-25
	4	0	20	30	15	25	15	10	20	15
	5	50	75	70	75	80	95	120	165	315
	6	-70	-65	-100	-145	-100	-165	-280	-2175 (191)a	-
	7	-35	-40	-10	-35	-35	-40	-35	-65	-40
	8	-	-	-	-	-	-	-	-	-
3	9	85	105	105	90	95	40	55	75	165
	10	-80	-75	-85	-110	-150	-140	-300	-1305 (214)a	-
	11	40	65	50	40	55	65	50	85	120
	12	-35	-25	-25	-45	-60	-60	-50	-50	-60

TABLE 8 (Continued). WEAR READINGS ON TENSION SIDE

(INCHES x 10^5)

RUN NO.	BAY NO.	6	24	50	72	100	137	175	250	350
	1	15	30	25	25	35	40	60	110	150
	2	70	105	110	100	120	125	110	135	155
	3	-5	-15	-5	-20	-25	-35	-35	-20	-40
4	4	30	35	45	40	20	15	35	25	25
	5	0	0	0	5	15	55	-210	-200	-990
	6	65	190	385	530	740	840	920	1175	1750
	7	-45	-40	-40	-45	-45	-50	-70	-80	-95
	8	-50	-50	-50	-70	-40	-55	-55	-35	-35
	9	50	185	525	765	1030	1420	2880 (168)[a]	-	-
4	10	-30	-10	5	-5	-10	-20	0	110	95
	11	35	95	285	470	635	825	985	1025 (193)[a]	-
	12	-40	-50	-45	-70	-85	-115	-150	-110	-800 (271)[a]

a Note: Testing of bearings 1-3, 1-7A, 2-2, 2-11, 3-6, 3-10, 4-9, 4-11, and 4-12 was discontinued after the test hours noted within the respective parentheses.

TABLE 9. TOTAL WEAR[b]

(RUNS #1, 2, 3, & 4) [INCHES x 10^5]

RUN NO.	BAY NO.	Time in Hours								
		6	24	50	72	100	137	175	250	361
	1	–	0	25	30	25	5	-5	35	15
	2	5	-5	10	55	225	355	470	805	2955
1	3	10	5	0	130	290	570	855	1760	3280 (325)a
	4	–	45	30	45	45	50	55	50	60
	5	–	40	55	100	100	100	115	155	170
	6	50	60	60	60	65	105	110	135	155
	7A*	15	50	145	460	835	1175	1645	2400 (189)a	
	8	–	45	55	70	70	105	95	115	125
1	9	–	75	65	75	95	140	155	155	235
	10	–	35	-20	-15	25	60	70	50	50
	11	20	15	-5	15	25	35	35	155	230
	12	30	5	20	65	70	75	115	140	155
		Time in Hours								
		11	24	47	72	102	137	158		
	7B*	70	165	360	555	775	1090	1165		
		Time in Hours								
		6	24	56	72	100	136	175	250	350
	1	60	75	70	75	85	90	85	65	65
	2	20	15	50	215	390	515	830	1105 (244)a	–
2	3	-25	-5	80	15	-5	-20	-45	-5	-20
	4	30	60	80	115	95	80	40	85	100
	5	60	55	45	50	10	35	60	95	85
	6	5	20	25	40	5	-25	-5	15	20

*Early failure in bay 7 allowed a second run. The two runs are identified as 7A and 7B.

TABLE 9 (Continued). TOTAL WEAR[b]

(RUNS #1, 2, 3, & 4) [INCHES x 10^5]

RUN NO.	BAY NO.	Time in Hours 6	24	56	72	100	136	175	250	350
	7	15	20	35	20	20	25	10	25	25
	8	45	30	50	10	5	65	30	25	35
	9	0	30	15	0	30	5	65	65	120
2	10	60	65	75	70	95	115	105	140	140
	11	20	10	185	190	190	155	295	380	260 (267)[a]
	12	25	15	40	75	65	40	-25	55	95

RUN NO.	BAY NO.	Time in Hours 6	24	50	72	100	137	175	250	350
	1	15	10	20	10	25	30	25	5	35
	2	20	35	35	45	35	35	35	20	10
	3	0	10	5	-10	0	-10	-10	-30	10
3	4	0	20	30	15	25	15	10	20	15
	5	50	75	70	75	80	95	120	165	315
	6	55	110	225	315	630	940	1310	2745 (191)[a]	
	7	60	80	75	75	90	105	105	110	115
	8	60	60	85	100	95	100	95	135	160
	9	-10	40	40	25	65	65	80	115	245
3	10	50	70	90	140	175	305	565	1880 (214)[a]	-
	11	40	65	50	40	55	65	50	85	120
	12	55	65	80	90	90	100	100	105	145

TABLE 9 (Continued). TOTAL WEAR[b]

(RUNS #1, 2, 3, & 4)[INCHES x 10^5]

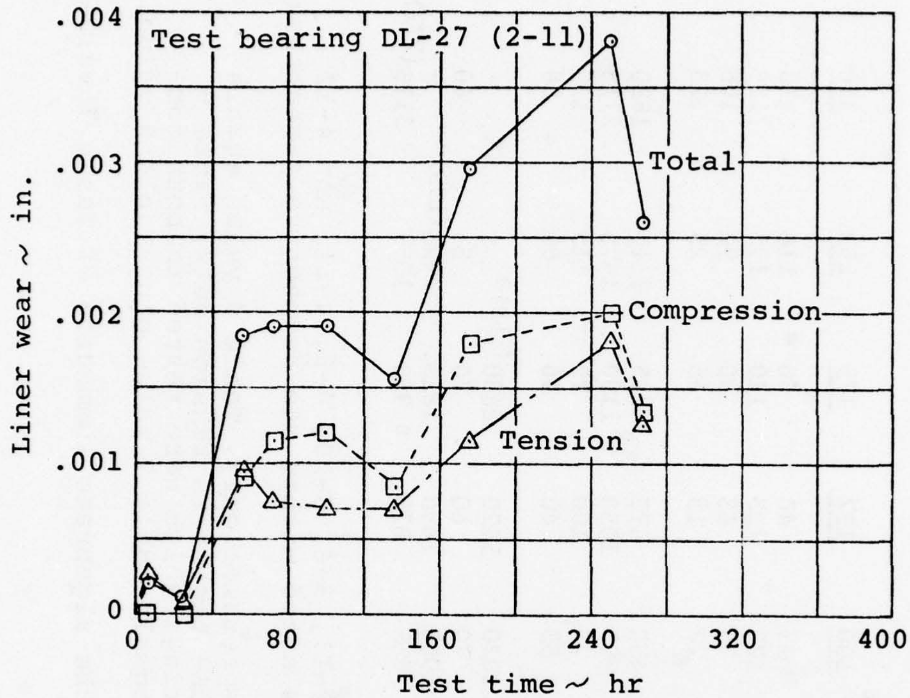
RUN NO.	BAY NO.	6.0	24.0	50.0	72.0	100	137	175	250	350
		Time in Hours								
	1	15	30	25	25	35	40	60	110	150
4	2	70	105	110	100	120	125	110	135	155
	3	15	45	60	70	75	65	80	90	105
	4	30	35	45	40	20	15	35	25	25
	5	5	30	215	400	450	475	725	1115	1880
4	6	-35	170	430	600	880	1050	1190	1475	2115
	7	60	70	80	65	70	100	75	115	130
	8	15	30	35	55	55	40	40	80	80
	9	50	185	525	765	1030	1420	2880 (168)[a]	-	-
4	10	65	75	80	80	70	60	70	55	60
	11	45	225	515	800	1085	1480	1815	1925 (193)[a]	-
	12	95	170	275	440	665	875	990	1355	3115 (271)[a]

Notes:

a. Testing of bearings 1-3, 1-7A, 2-2, 2-11, 3-6, 3-10, 4-9, 4-11, and 4-12 was discontinued after the test hours noted within the respective parentheses.

b. For bearings which were tested in compression only, Table 9 value equals Table 7 value. For bearings which were tested in tension only, Table 9 value equals Table 8 value. For bearings which were tested in both compression and tension, Table 9 value equals Table 7 value plus Table 8 value.

c. For all bearings radial play equals the algebraic summation of Table 7 values and Table 8 values.



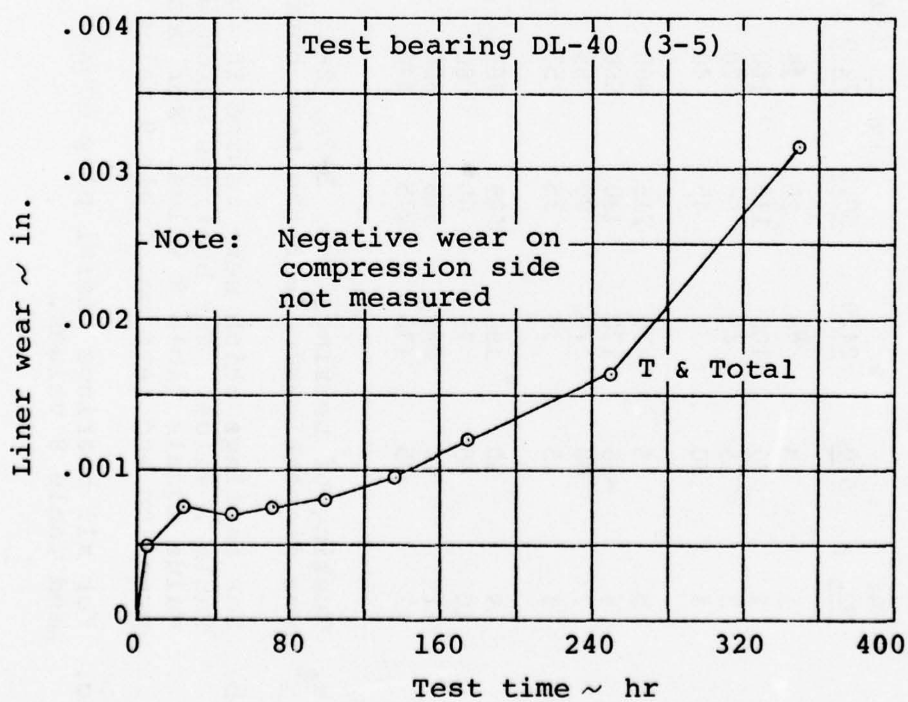


Figure 9. Liner wear vs test time - test bearings DL-27 (2-11) and DL-40 (3-5).

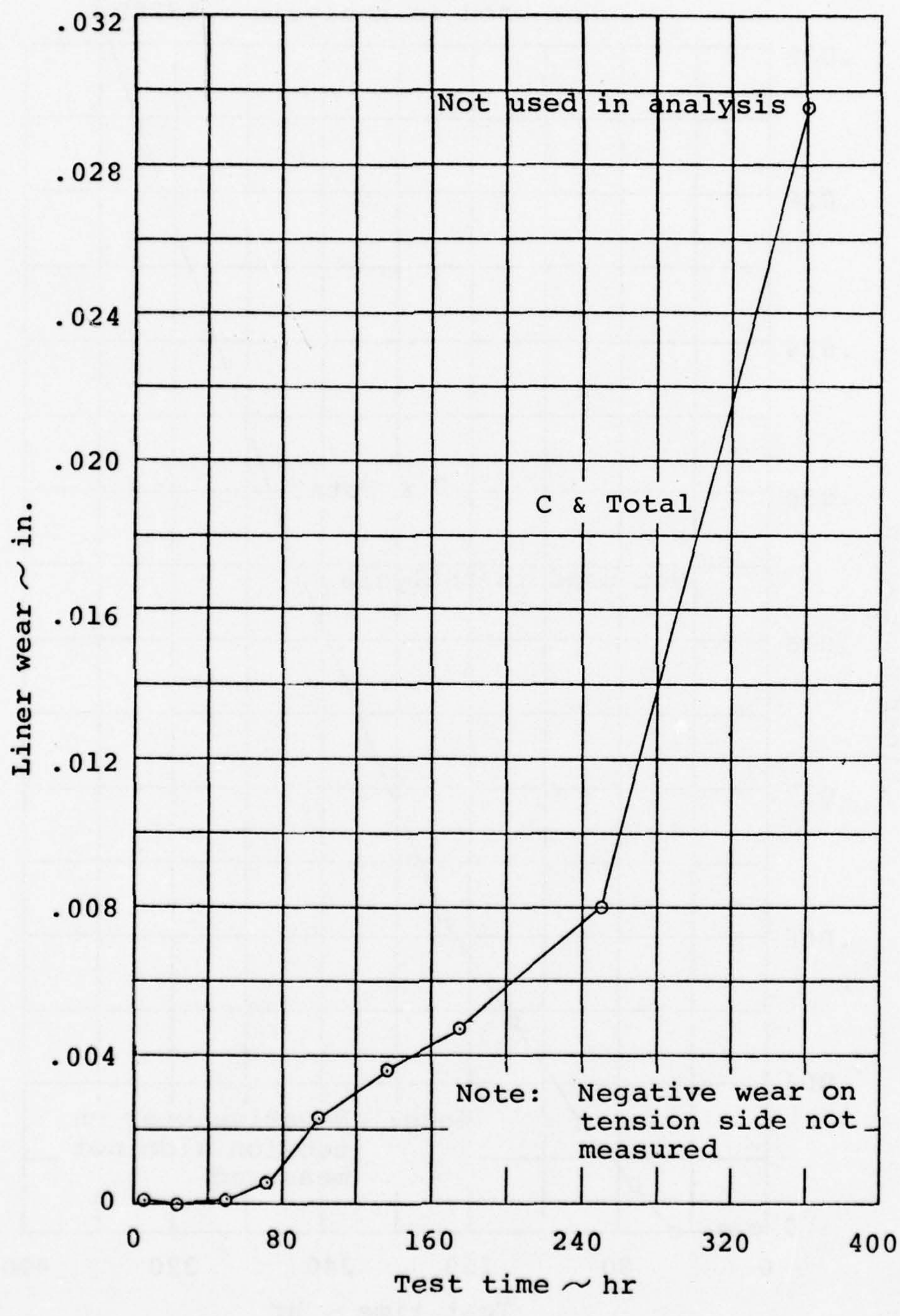


Figure 10. Liner wear vs test time - test bearing DL-14 (1-2).

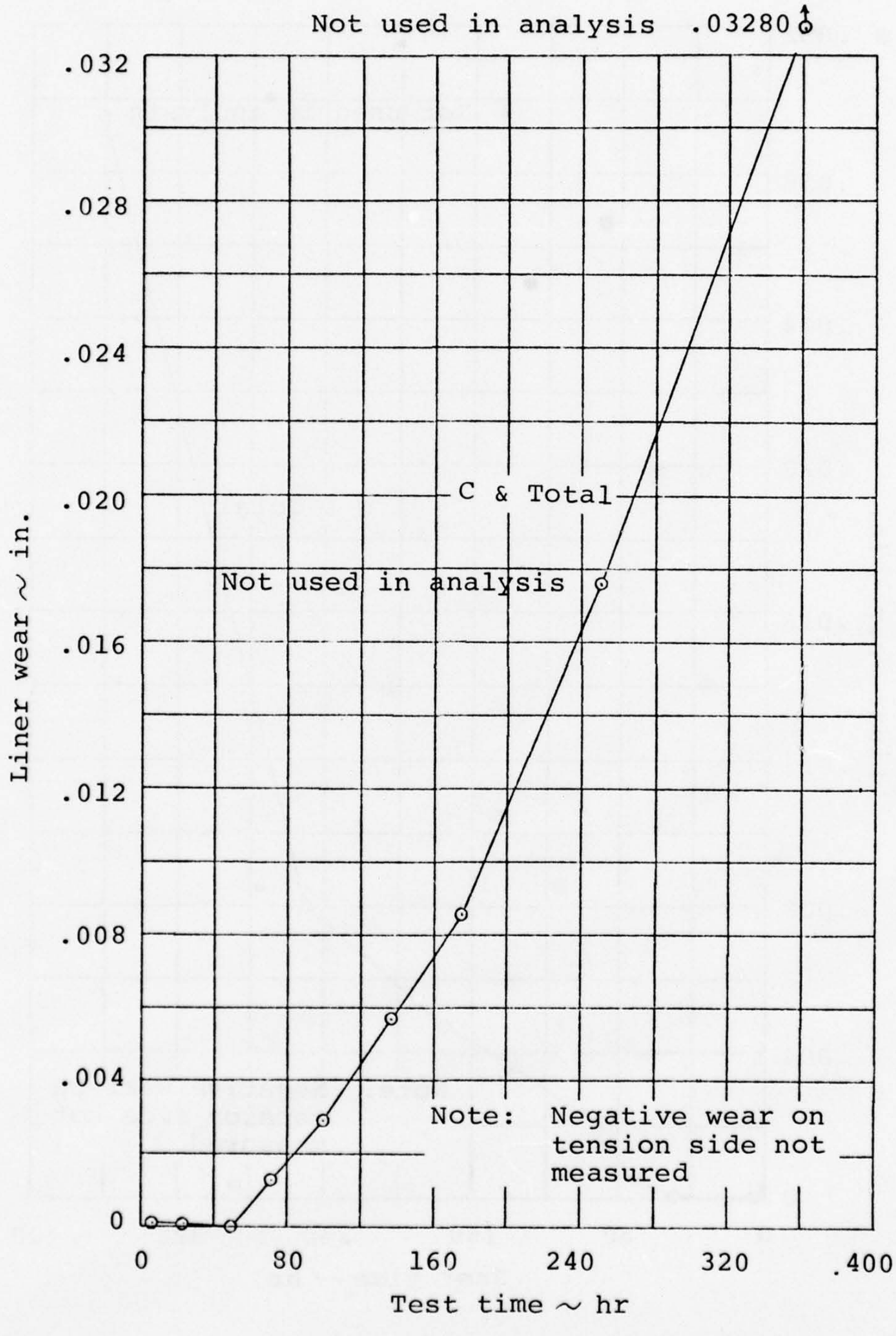


Figure 11. Liner wear vs test time - test bearing DL-11 (1-3).

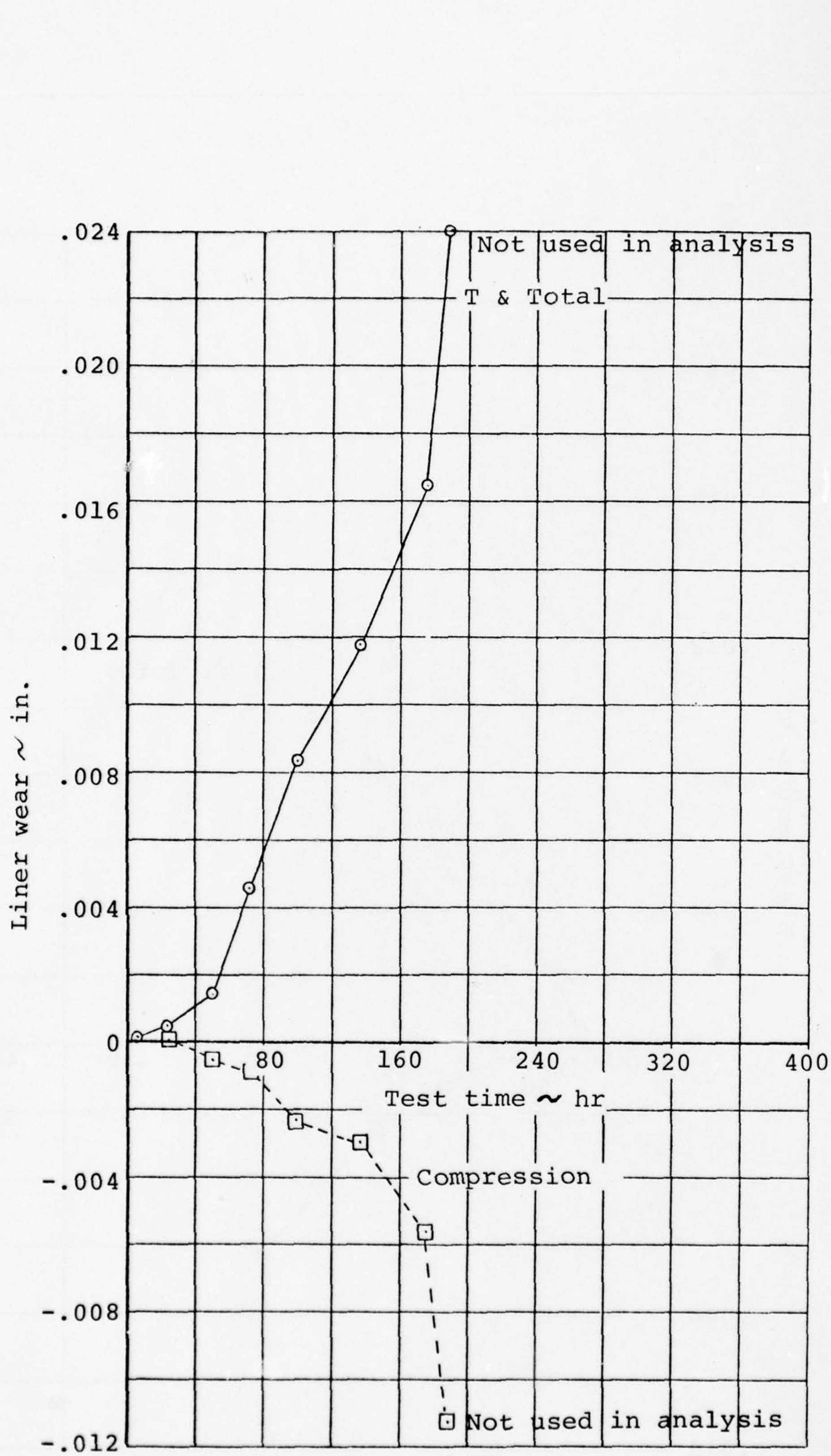


Figure 12. Liner wear vs test time - test bearing DL-7 (1-7A).

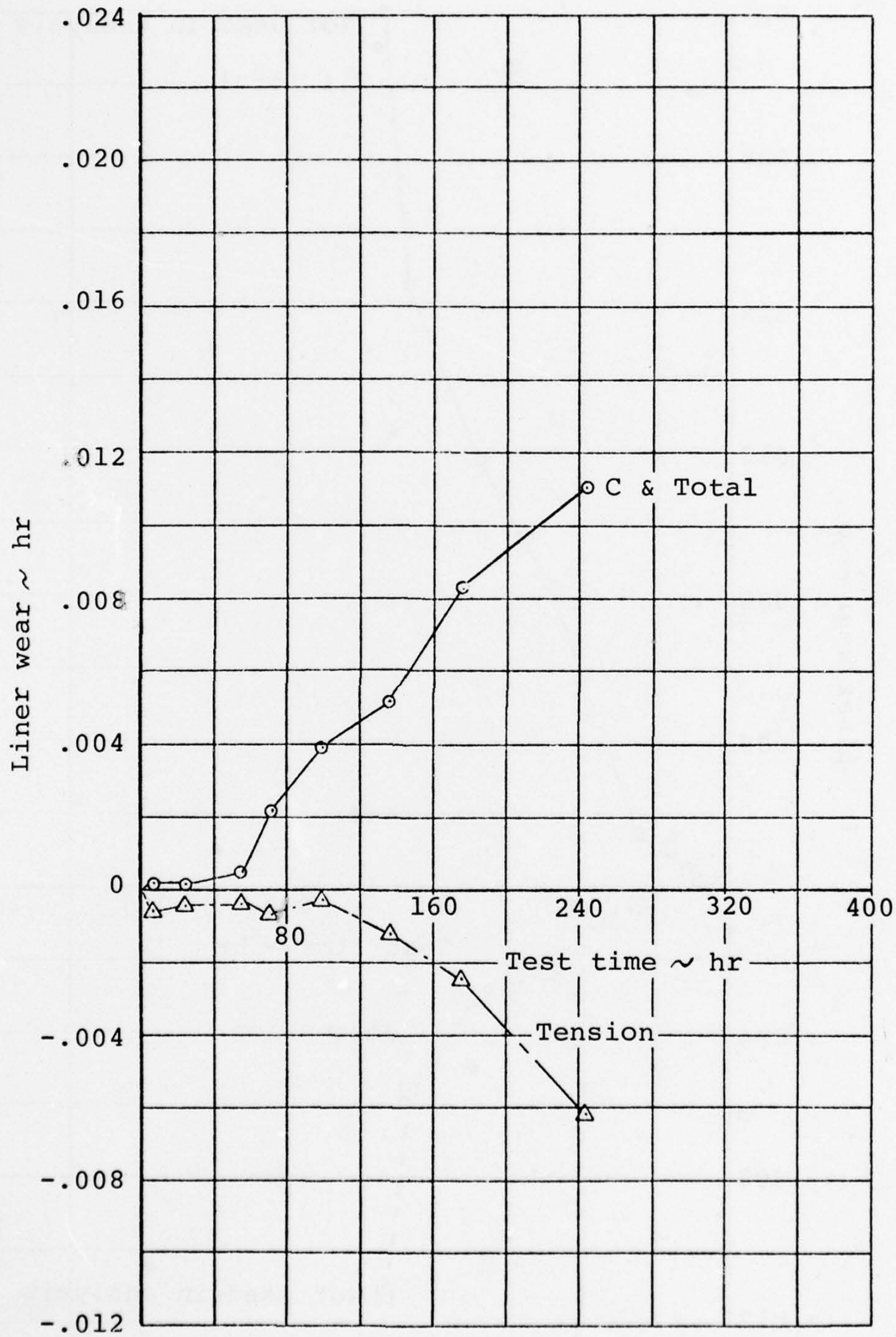


Figure 13. Liner wear vs test time - test bearing DL-26 (2-2).

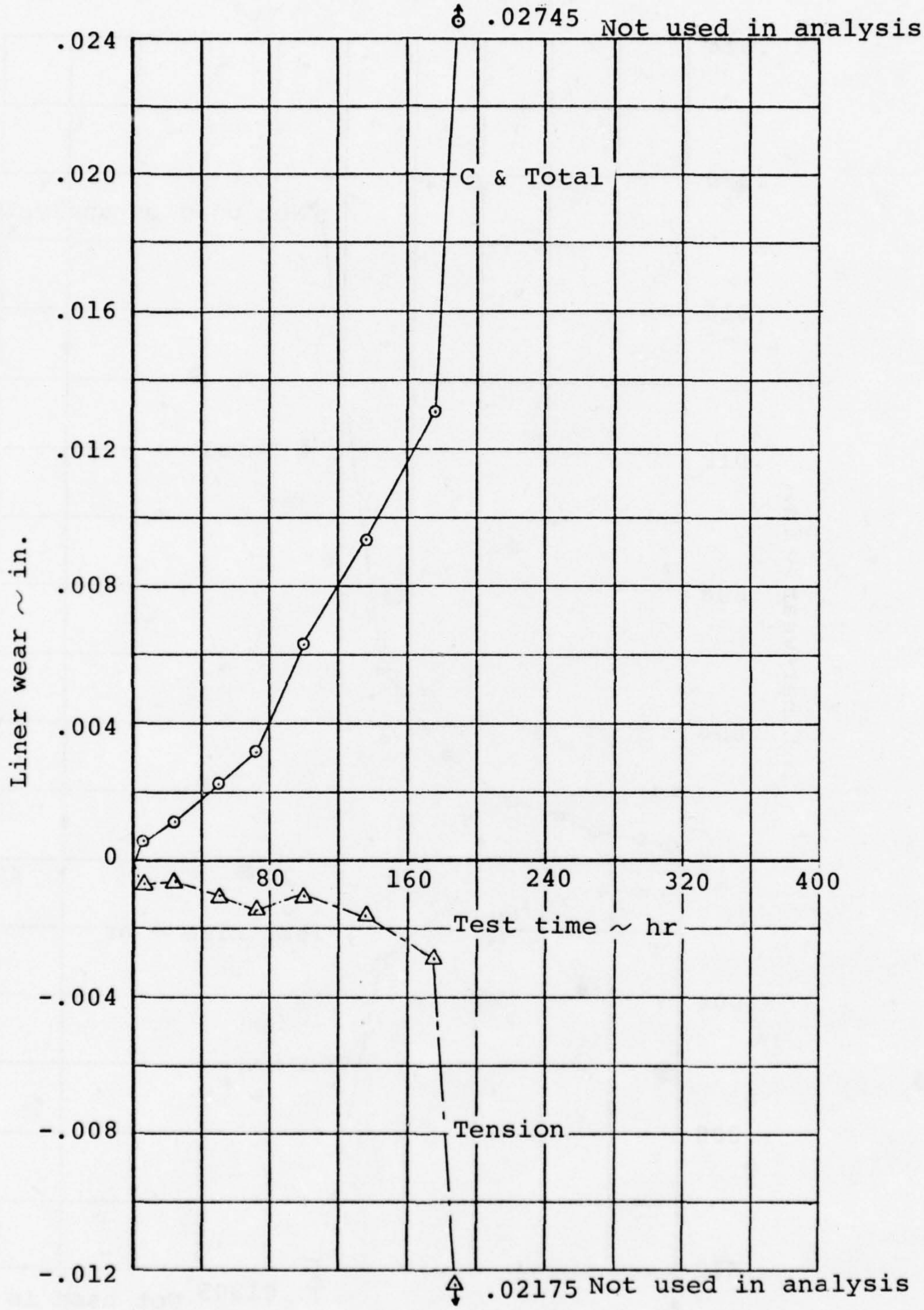


Figure 14. Liner wear vs test time - test bearing DL-41 (3-6).

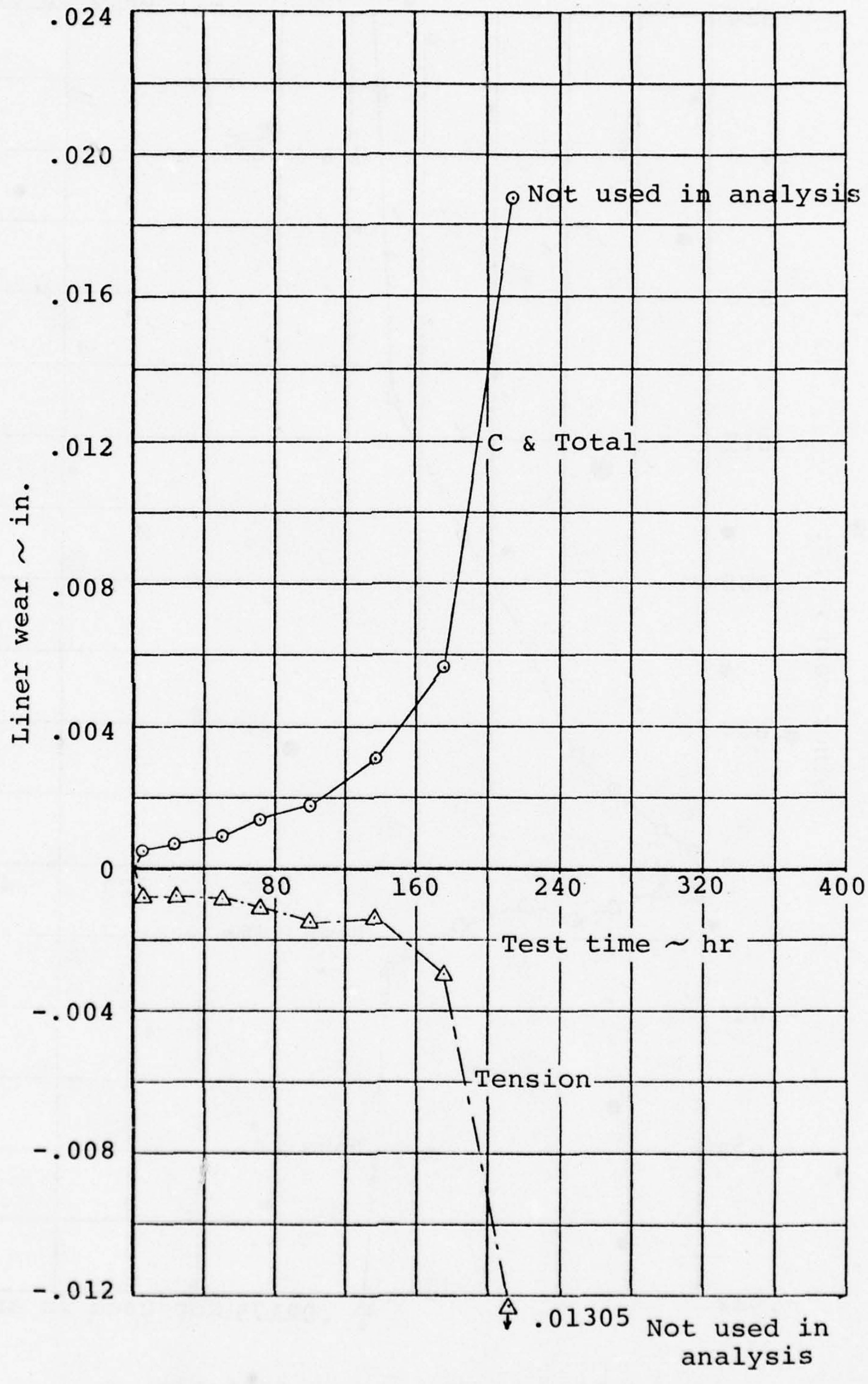


Figure 15. Liner wear vs test time - test bearing DL-43 (3-10).

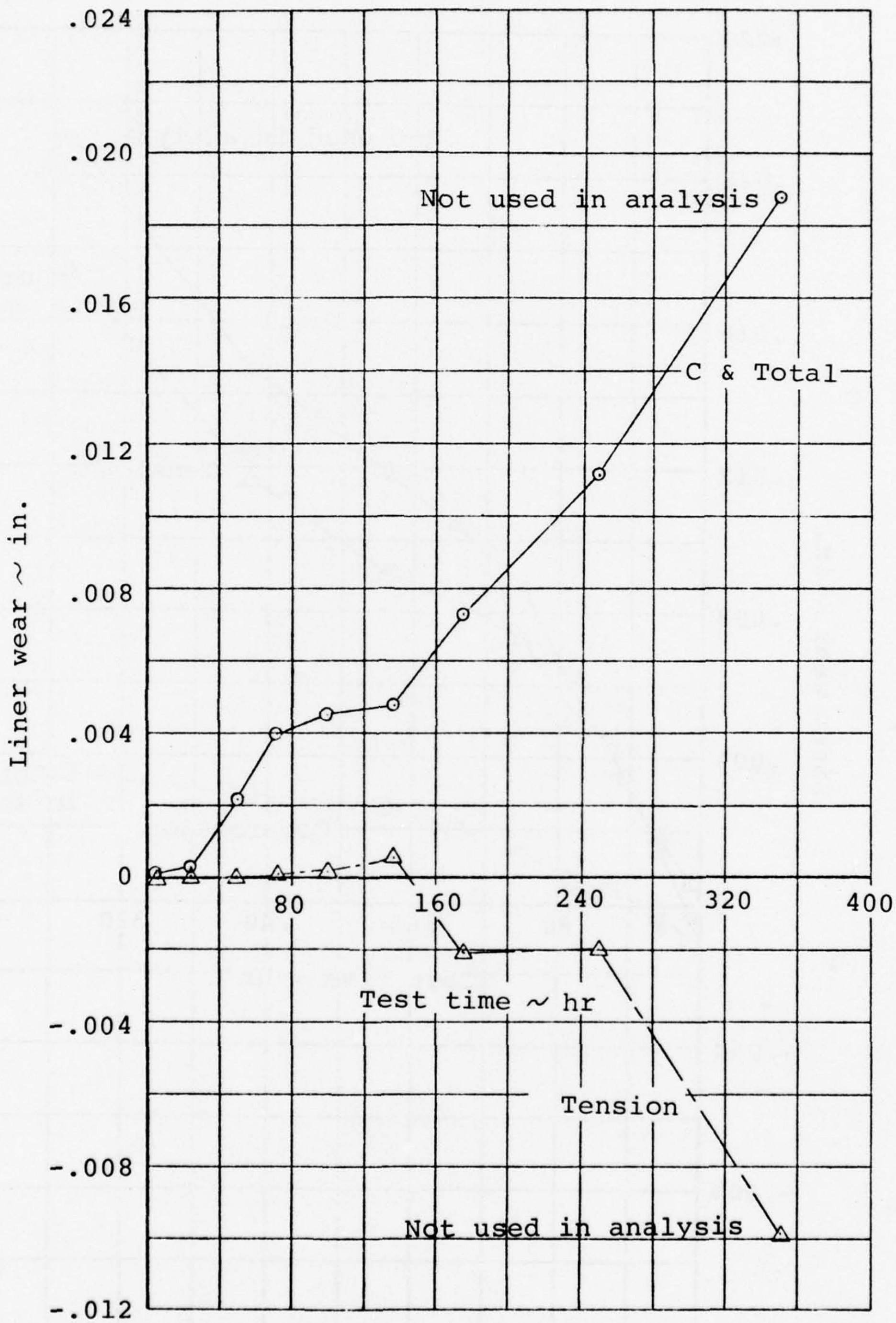


Figure 16. Liner wear vs test time - test bearing DL-49 (4-5).

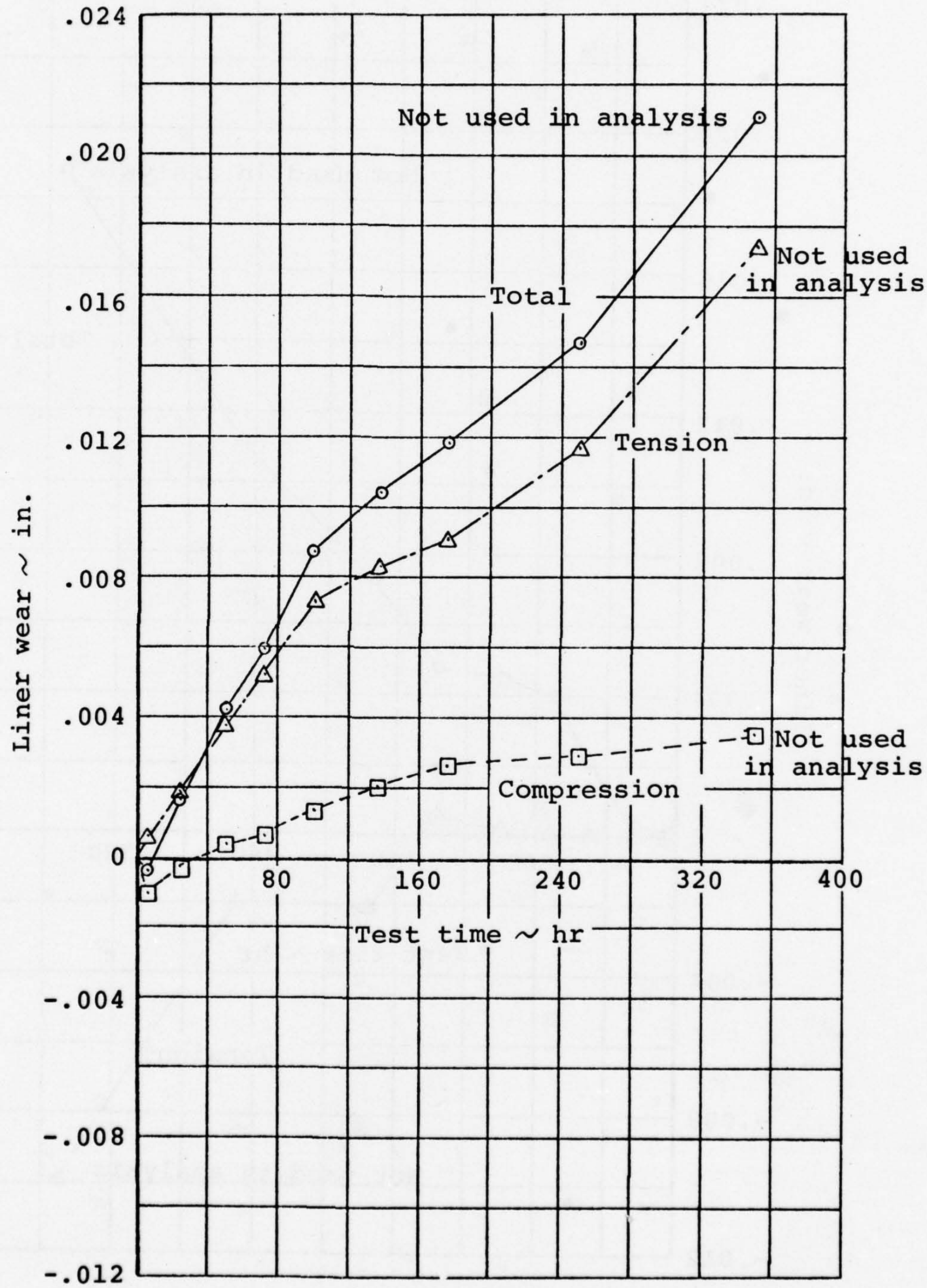


Figure 17. Liner wear vs test time - test bearing DL-50 (4-6).

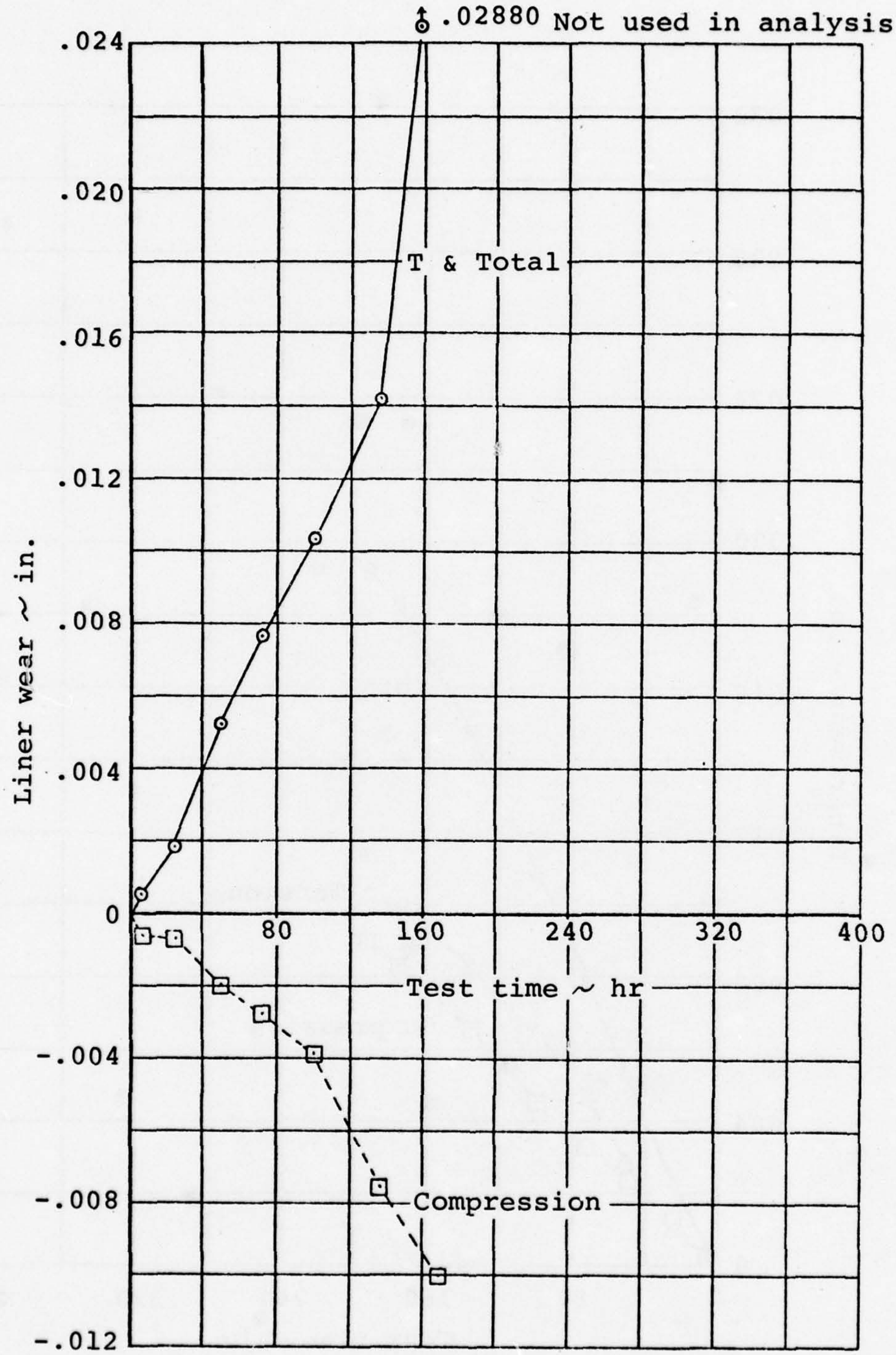


Figure 18. Liner wear vs test time - test bearing DL-54 (4-9).

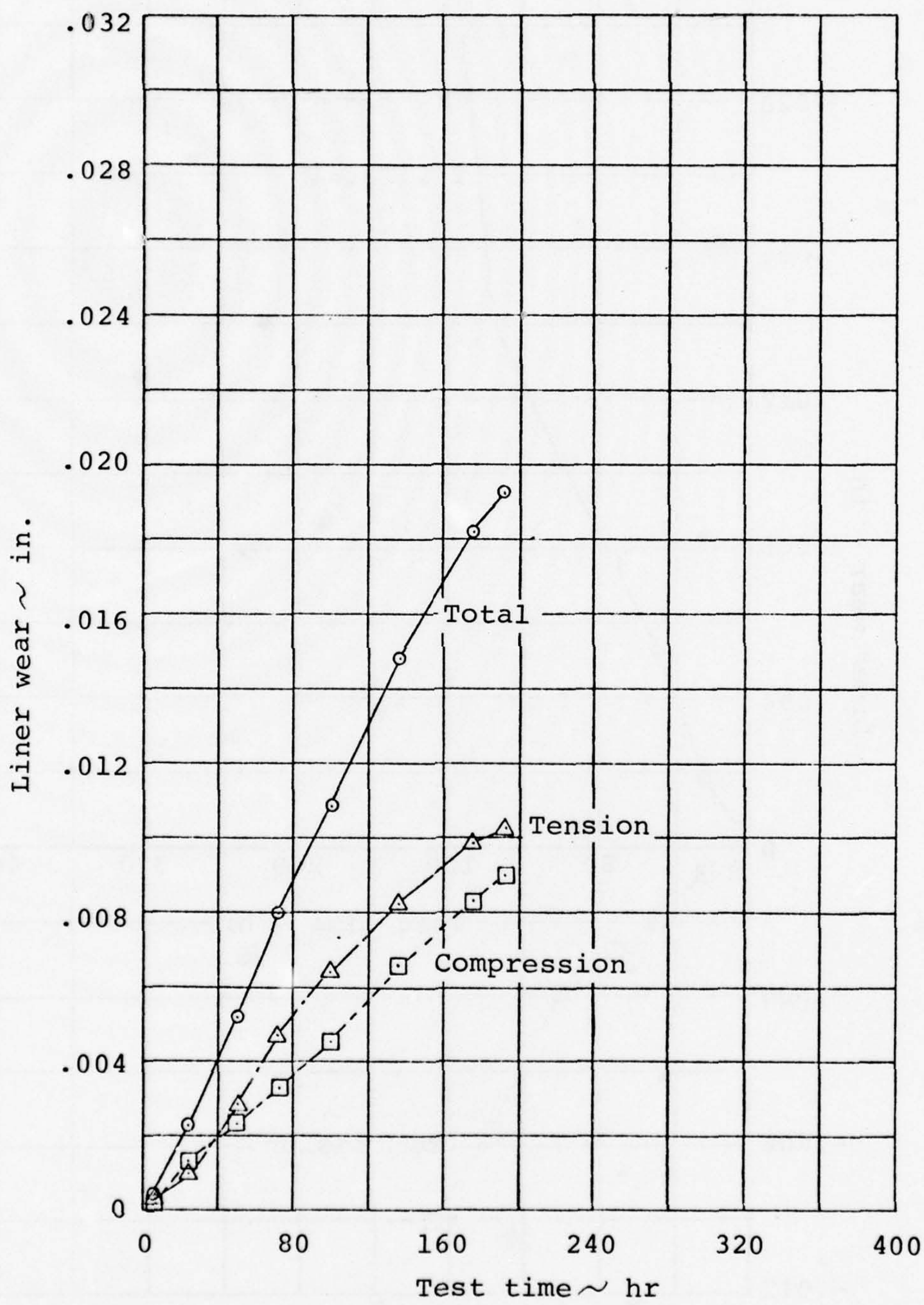


Figure 19. Liner wear vs test time - test bearing DL-53 (4-11).

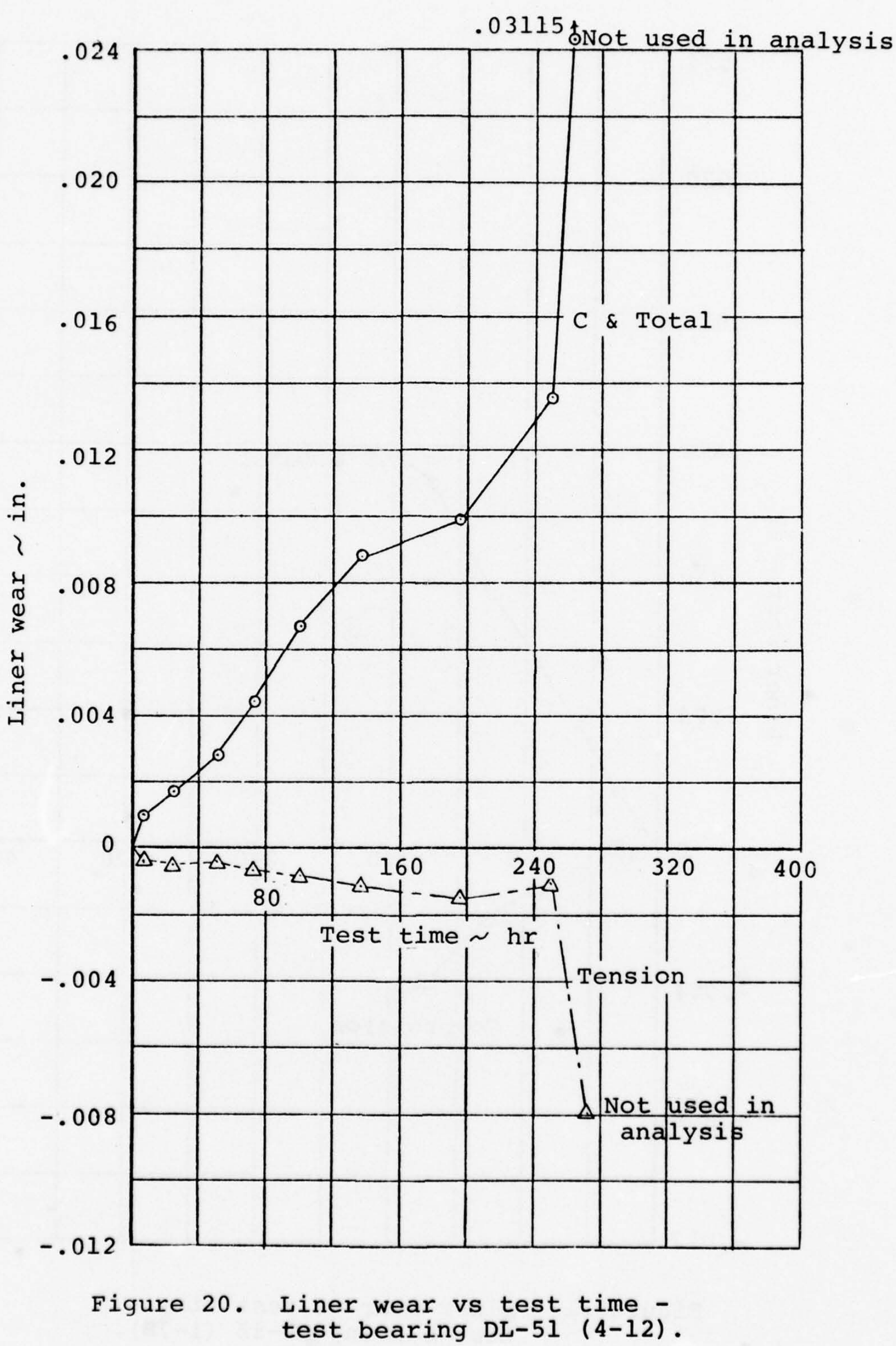


Figure 20. Liner wear vs test time - test bearing DL-51 (4-12).

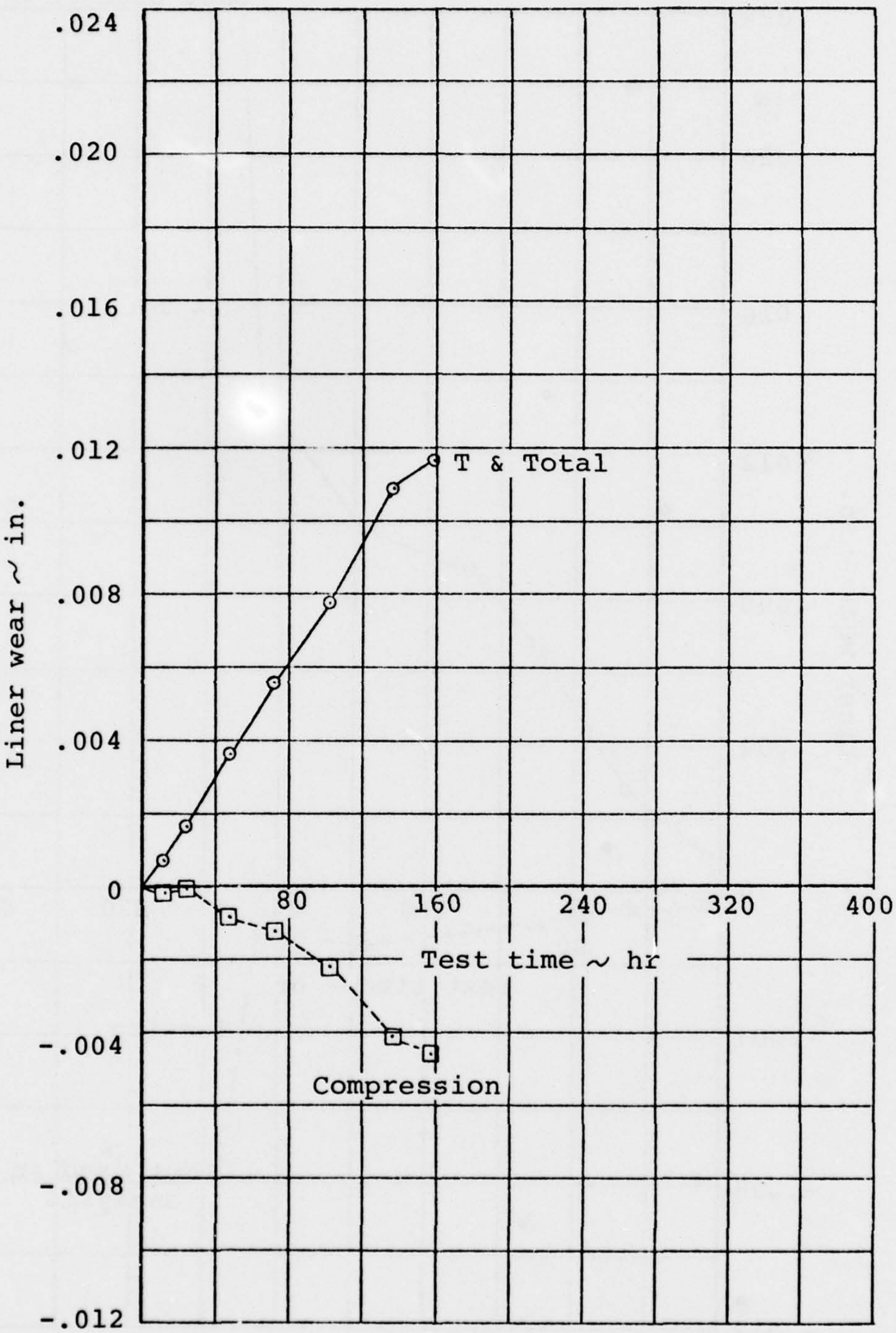


Figure 21. Liner wear vs test time - test bearing DL-15 (1-7B).

REPLICATION ON SCREENING TEST NUMBER 3 (TWELVE-BAY TEST RIG)

Test bearing DL-7 (1-7A) failed after only one-half of the planned 350-hour test run had been completed. In order to gain information on repeatability, test bearing DL-15 (1-7B) was installed and tested at the same conditions as DL-7 (1-7A). Figure 21 presents a plot of liner wear versus time for DL-15 (1-7B). Figure 12 is the corresponding plot for DL-7 (1-7A). Table 10 presents a comparison of these two bearings. Note that the coefficient of variation is high in the low wear regime of 6 to 50 hours, but reduces to approximately 7 to 17% in the high wear regime. These values of coefficient of variation indicate the variability in wear measurement caused by inherent differences in test bearings, the ability to test the differences, the ability to measure the differences, the ability to duplicate the test conditions, the repeatability of the water contamination-ball corrosion phenomenon, etc.

Also shown in Table 10 are coefficients of variation obtained from the 48 test bearings tested in the twelve-bay rig. On the average these values of coefficient of variation exceed the values obtained from DL-7 and DL-15 by 105%. Thus, it can be concluded that the test conditions used for the 48 test bearings have produced significant differences in wear.

WEAR SLOPES AND BREAK-IN WEAR

The following presentation is introduced for two reasons:

1. To illustrate that the trend of the average wear data increases and is linear with time.

2. To introduce the concept of break-in wear.

The total wear for each of the 48 bearings was summed and the total divided by 48 to obtain the mean (average) of total bearing wear for each of the measuring times as shown in Table 11. This was also done for the 36 bearings with no water contamination and the 12 bearings with water contamination. A least-squares fit of the data in each case shows correlation coefficients of 0.997 to 0.973 indicating extremely good linearity with time. These three plots are presented in Figure 22. Note that the three straight lines intersect at approximately 25 hours of test time regardless of the lack of or existence of water contamination. The 25-hour time may well be the average time required for water to invade the interior of the test bearings, start the ball corrosion mechanism, and generate sufficient abrasive corrosion product for a dominant wear process.

TABLE 10. COMPARISON OF TEST BEARINGS DL-7 AND DL-15

TIME (HOURS)	WEAR (INCHES X 100,000) DL-7	WEAR (INCHES X 100,000) DL-15	MEAN $\overline{X}$	RANGE R	σ_{EST} (NOTE 3)	COEFFICIENT OF VARIATION C_V (%) (NOTE 4)	COEFFICIENT OF VARIATION FOR 48 BRGS. (%) (NOTE 5)	ΔC_V (%)
6	15	25I[1]	20	10	8.9	44.5	----	----
24	50	165	107.5	115	102.0	94.9	91.7	-3.2
50	145	370I[1]	257.5	225	199.5	77.5	129.5	52.0
72	460	555	507.5	95	84.2	16.6	141.5	124.9
100	835	770I[1]	802.5	65	57.6	7.2	150.7	143.5
137	1175	1090	1132.5	85	75.4	6.7	158.5	151.8
175	1645	1360E[2]	1502.5	285	252.7	16.8	176.6	159.8

Notes:

1. I means interpolated values (Wear for DL-15 measured at 11, 24, 27, 72, 102, 137, and 158 hours).
2. E means extrapolated value (Last wear reading for DL-15 was at 158 hours).
3. $\sigma_{EST} = \frac{\overline{R}}{d_2} \quad \frac{R_{SS_2}}{1.128}$.
4. These values for coefficient of variation indicate the variability in wear measurement caused by inherent differences in test bearings, measuring accuracies, load duplication, and all other ambient conditions which change as the test proceeds.
5. These values for coefficient of variation were copied from Table 11 in order to show that the test conditions used for the 48 bearings have caused significant variations in the wear results.

TABLE 11. TABULATION OF BEARING TOTAL WEAR AVERAGES

48 TEST BEARINGS (WITH & W/O WATER)

$\bar{x}$ TIME (HR)	$\bar{y}$ AVE. OF TOTAL BRG. WEAR (INx10^{-5})	σ	Cv(%)
24	53.54	49.07	91.7
50	91.46	118.46	129.5
72	131.77	186.46	141.5
100	182.81	275.55	150.7
137	238.44	377.99	158.5
175	327.40	578.03	176.6

$\bar{\bar{x}} = 93$ hr $\bar{\bar{y}} = 170.90\text{x}10^{-5}$ in.

$\sigma_{\bar{x}} = 56.14$ hr $\sigma_{\bar{y}} = 100.81\text{x}10^{-5}$ in.

$r = 0.997$

$y = 1.79084(x) + 4.35533$

36 TEST BEARINGS (W/O WATER)

$\bar{x}$	$\bar{y}$	σ	Cv(%)
24	42.64	25.99	61.0
50	47.92	28.02	58.5
72	50.42	33.01	65.5
100	53.61	33.46	62.4
137	59.17	41.79	70.6
175	59.58	45.69	76.7

$\bar{\bar{x}} = 93$ hr $\bar{\bar{y}} = 52.22\text{x}10^{-5}$ in.

$\sigma_{\bar{x}} = 56.14$ hr $\sigma_{\bar{y}} = 6.602\text{x}10^{-5}$ in.

$r = 0.973$

$y = 0.114389(x) + 41.5852$

12 TEST BEARINGS (WITH WATER)

$\bar{x}$	$\bar{y}$	σ	Cv(%)
24	86.25	78.61	91.1
50	222.08	176.14	79.3
72	375.83	237.43	63.2
100	570.42	316.28	55.4
137	776.25	424.97	54.7
175	1130.83	685.22	60.6

$\bar{\bar{x}} = 93$ hr $\bar{\bar{y}} = 526.94\text{x}10^{-5}$ in.

$\sigma_{\bar{x}} = 56.14$ hr $\sigma_{\bar{y}} = 384.37\text{x}10^{-5}$ in.

$r = 0.996$

$y = 6.81998(x) - 107.315$

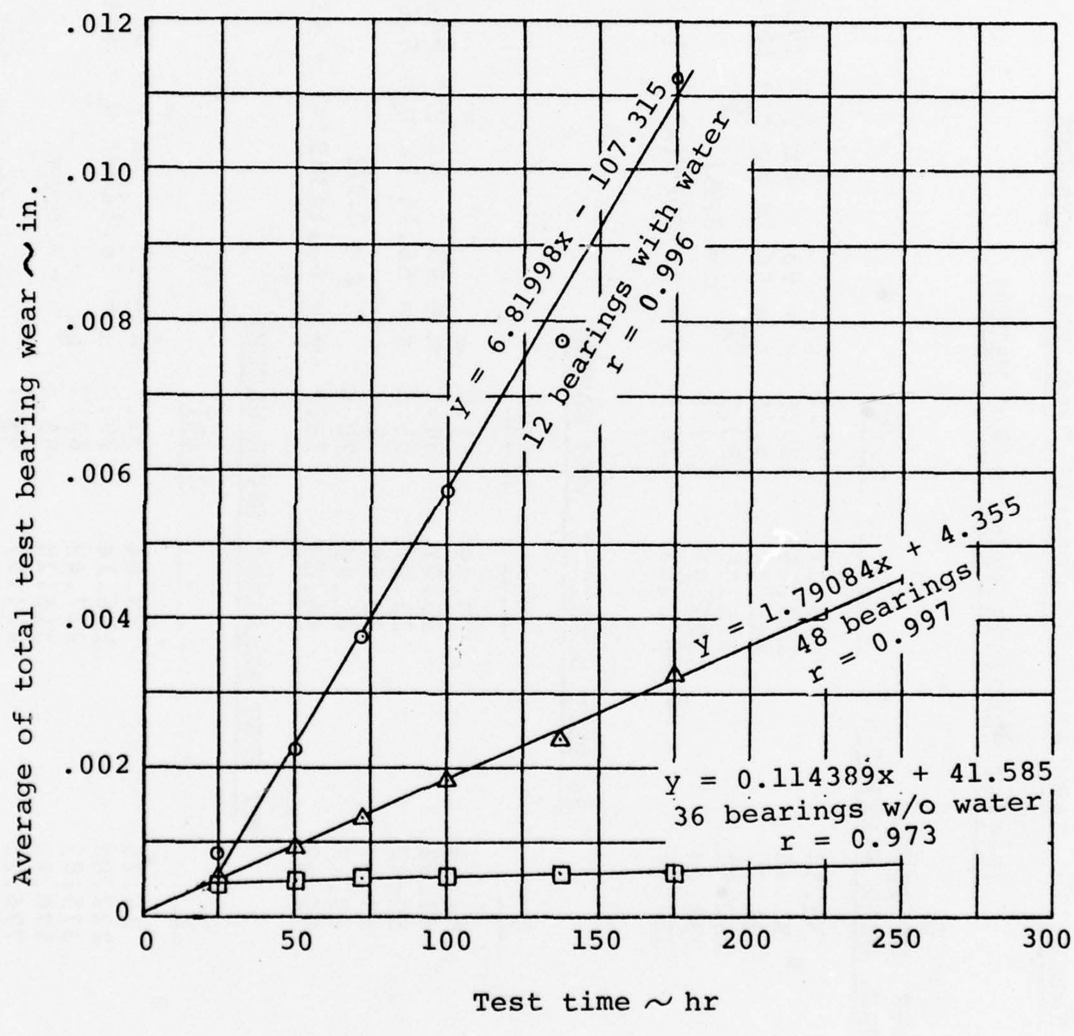


Figure 22. Average of total test bearing wear vs test time.

Because it was found that the average of the total bearing wear is linear with time and the existence of the break-in wear at 25 hours can be thought of as an intercept, it was decided to obtain an equation for the slope of the wear curves. (Note the analogy between this concept and the equation for a straight line which includes an intercept and a slope.) Therefore, preliminary plots were made for all 48 test bearings. These plots were used to determine data points which were obviously inconsistent with the majority of the data. Data points obtained after the bearing failed were definitely in this category. A least-squares analysis was done for each of the 48 bearings to obtain the slopes of the wear curves. Table 12 presents the slopes so obtained with the intercepts and correlation coefficients added as additional information. It can be noted from Figure 23, which is a plot of slope versus correlation coefficient, that the bearings with correlation coefficients less than 0.90 are virtually zero wear situations.

WEAR TEST RESULTS (CONTINUOUS ROTATION TEST RIG)

Ten bearings were tested in the continuous rotation test rig. Table 13 lists the wear values (both radial and axial) obtained. Figures 24, 25, and 26 are plots of radial wear versus test time and Figure 27 is a plot of axial wear versus test time.

SELECTION OF TEST CONDITIONS FOR VALIDATION TESTS

After the first eight bearings were tested in the screening tests, an analysis of variance was performed on the slopes of the plots from the wear data. Table 14 tabulates the slope values and also presents the analysis of variance. As can be seen in the primary ANOVA table of Table 14, one of the first-order interactions has a high mean square. The presence of this high mean square value causes all four main effects to be nonsignificant at the .05 level of significance. The testing of eight additional bearings would normally be performed, thus revising the half-replicate Latin square design into a full replicate. However, not having the luxury of time or money, an unorthodox "pooling" of the variables was done as shown in Table 14. Thereupon, three sources of variation appeared to be significant at the .05 level of significance. Two of these were the main effects of static radial pressure and static axial load, each of which was confounded with the second-order interactions of static axial load X surface velocity X ball

TABLE 12. COMPILATION OF CORRELATION COEFFICIENT, SLOPE, AND INTERCEPT FOR 48 BEARINGS

BEARING DESIGNATION	r	SLOPE	INTERCEPT	BEARING DESIGNATION	r	SLOPE	INTERCEPT
DL-13(1-1)	.076	0.00991	14.80	DL-37(3-1)	.426	0.03820	14.50
DL-14(1-2)	.991	3.73917	-158.04	DL-32(3-2)	.614	-0.05923	37.66
DL-11(1-3)	.987	8.66109	-530.88	DL-38(3-3)	.246	-0.02764	-0.31
DL-12(1-4)	.773	0.06054	38.65	DL-39(3-4)	.424	-0.02467	22.32
DL-9 (1-5)	.940	0.36630	50.85	DL-40(3-5)	.938	0.67904	28.29
DL-10(1-6)	.970	0.32169	46.89	DL-41(3-6)	.985	7.59995	-100.20
DL-7 (1-7)	.995	11.7203	-399.73	DL-33(3-7)	.899	0.15207	70.89
DL-8 (1-8)	.926	0.23858	50.14	DL-34(3-8)	.953	0.27136	63.79
DL-1 (1-9)	.966	0.49574	51.93	DL-42(3-9)	.946	0.61283	-5.37
DL-3 (1-10)	.566	0.16751	7.40	DL-43(3-10)	.934	2.80131	-26.42
DL-5 (1-11)	.948	0.69820	-38.90	DL-35(3-11)	.862	0.19448	38.18
DL-6 (1-12)	.946	0.43143	18.67	DL-36(3-12)	.950	0.21756	64.08
DL-28(2-1)	.099	-0.00919	75.64	DL-52(4-1)	.967	0.39127	3.84
DL-26(2-2)	.971	4.97172	-112.75	DL-44(4-2)	.903	0.14859	98.49
DL-18(2-3)	.319	-0.10037	9.70	DL-45(4-3)	.939	0.15726	50.99
DL-19(2-4)	.360	0.08928	64.51	DL-46(4-4)	.410	-0.03515	34.55
DL-25(2-5)	.584	0.13106	37.98	DL-49(4-5)	.980	5.28199	-104.26
DL-20(2-6)	.072	-0.01209	12.68	DL-50(4-6)	.988	5.56143	183.73
DL-24(2-7	.124	0.00782	20.65	DL-55(4-7)	.901	0.19438	59.86
DL-22(2-8)	.066	0.01175	29.54	DL-47(4-8)	.851	0.16532	26.40
DL-21(2-9)	.889	0.31618	-4.40	DL-54(4-9)	.999	10.6289	-26.61
DL-23(2-10)	.944	0.26006	62.33	DL-48(4-10)	.659	-0.05257	75.13
DL-27(2-11)	.916	1.39635	35.17	DL-53(4-11)	.998	10.2930	12.27
DL-29(2-12)	.397	0.12556	26.47	DL-51(4-12)	.994	5.35076	63.69

NOTES:

1. r = Correlation coefficient
2. Slope in inches/hour x 10^5
3. Intercept in inches x 10^5

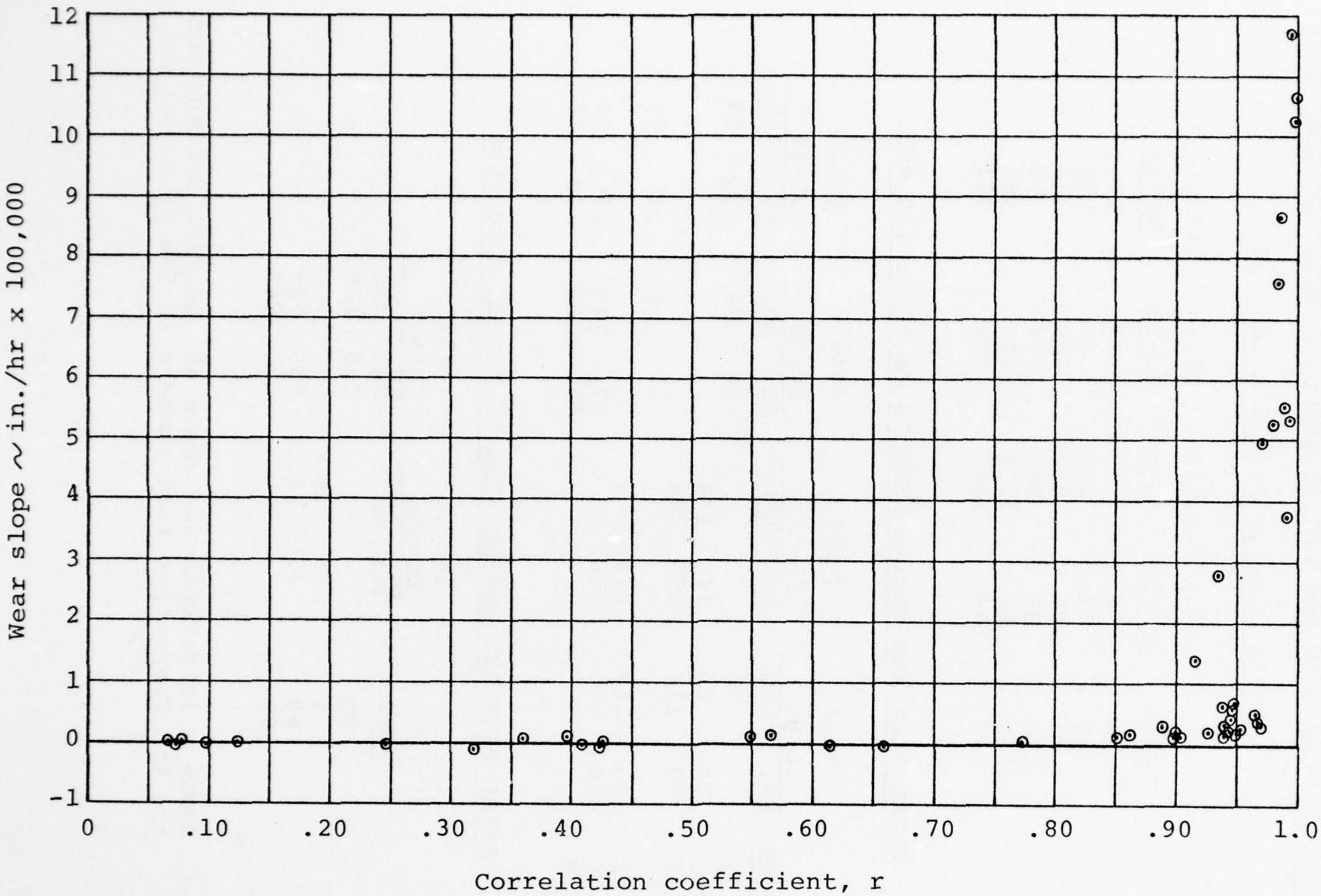


Figure 23. Wear slope vs correlation coefficient.

TABLE 13. WEAR TEST RESULTS (CONTINUOUS ROTATION TEST RIG)

RADIAL WEAR IN INCHES x 100,000

HOURS	(#50) DL-2	(#51) DL-4	(#53) DL-30	(#56) DL-31
30	64	119	104	124
60	74	249	219	194
90	104	504	459	324
120	89	846	454	359
160	89	1159	479	359
200	144	1364	604	444

AXIAL PLAY AND RADIAL WEAR IN INCHES x 100,000

	(#55) DL-56		(#54) DL-57		(#49) DL-58		(#52) DL-59	
HOURS	AX.	RAD.	AX.	RAD.	AX.	RAD.	AX.	RAD.
29					80	29	25	39
30	70	49	140	139				
58	90	114	195	204				
70					--	549	125	59
72	200	269	275	229				
96	410	479	560	549	375	944	125	64
122					580	1129	140	79

AXIAL PLAY AND RADIAL WEAR IN INCHES x 100,000

	(#57) DL-60		(#58) DL-61	
HOURS	AX.	RAD.	AX.	RAD.
30			280	239
34	85	34		
60	85	54	350	589
90			420	844

Note: Numbers in parentheses above the bearing designation are the respective test numbers in Tables 2, 6, and 15.

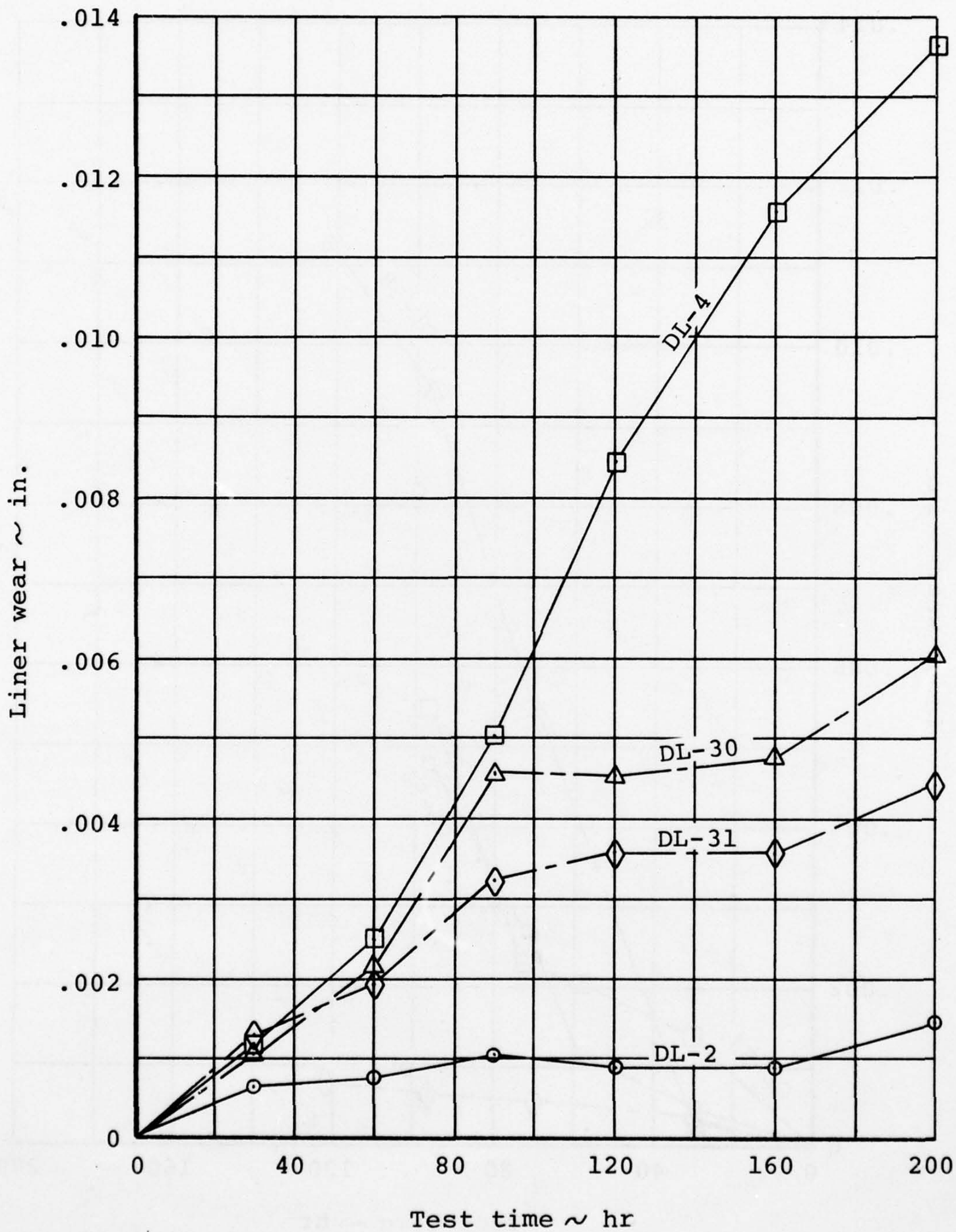


Figure 24. Liner wear vs test time - test bearings DL-2, DL-4, DL-30 and DL-31.

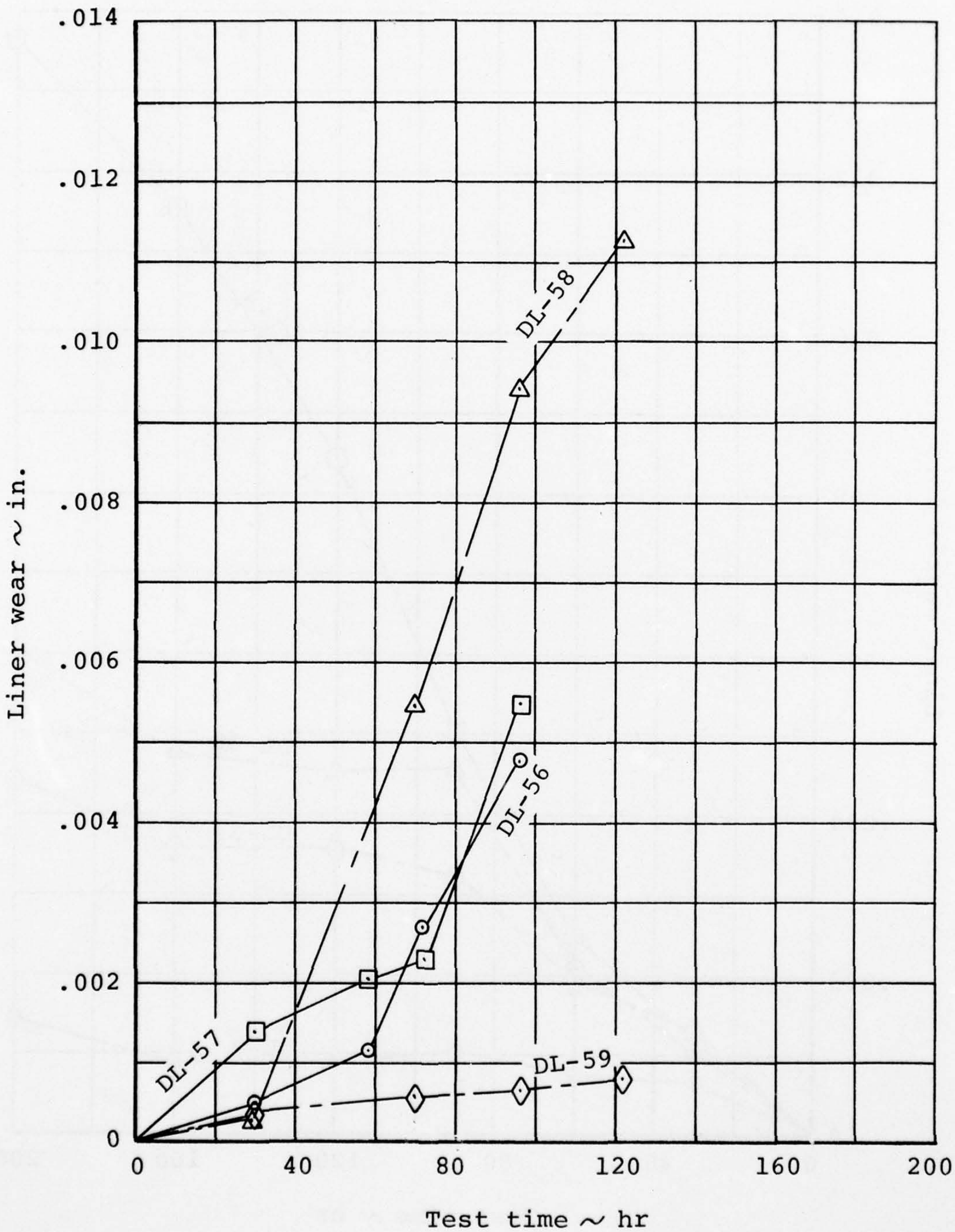


Figure 25. Liner wear vs test time - test bearings DL-56, DL-57, DL-58 and DL-59.

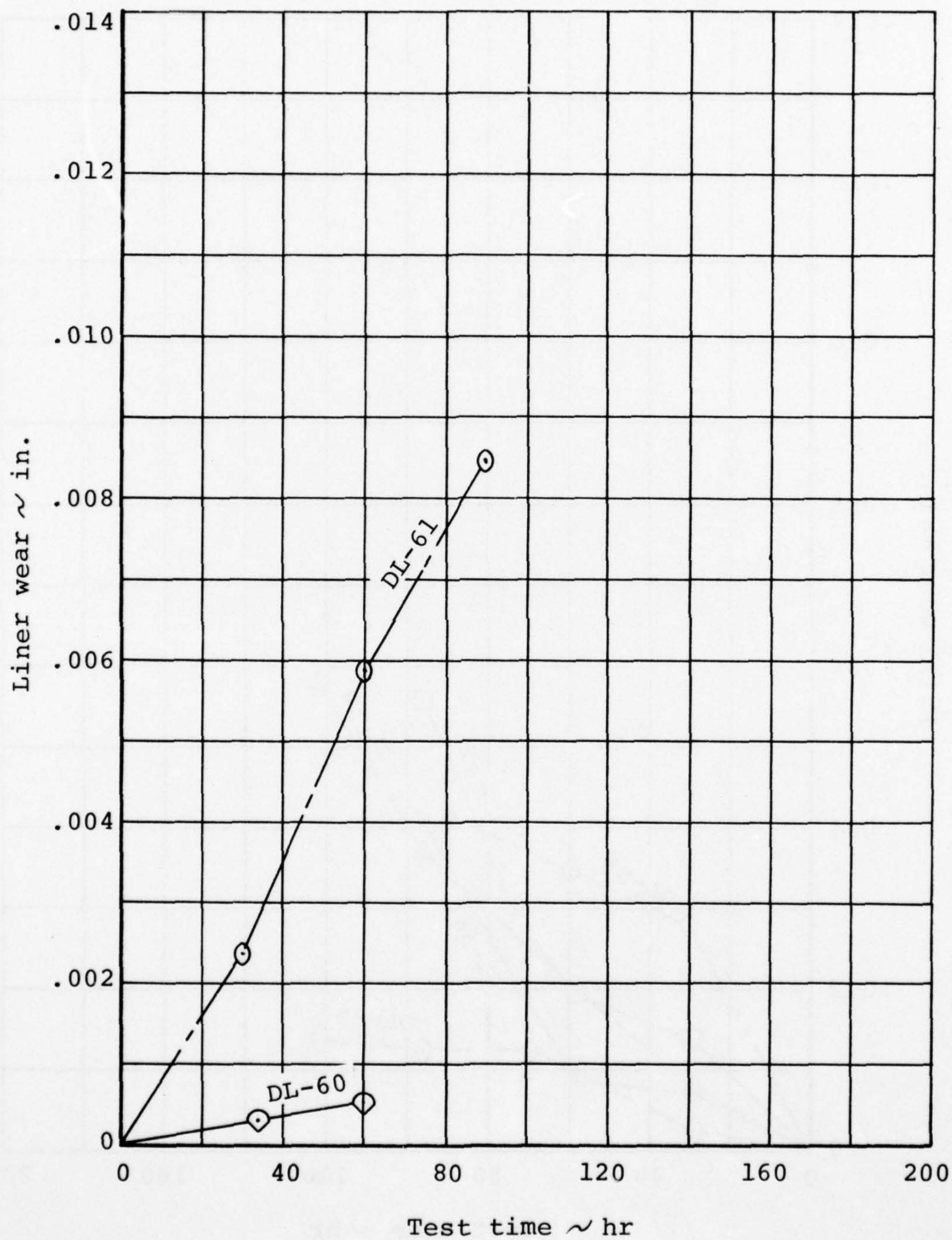


Figure 26. Liner wear vs test time - test bearings DL-60 and DL-61.

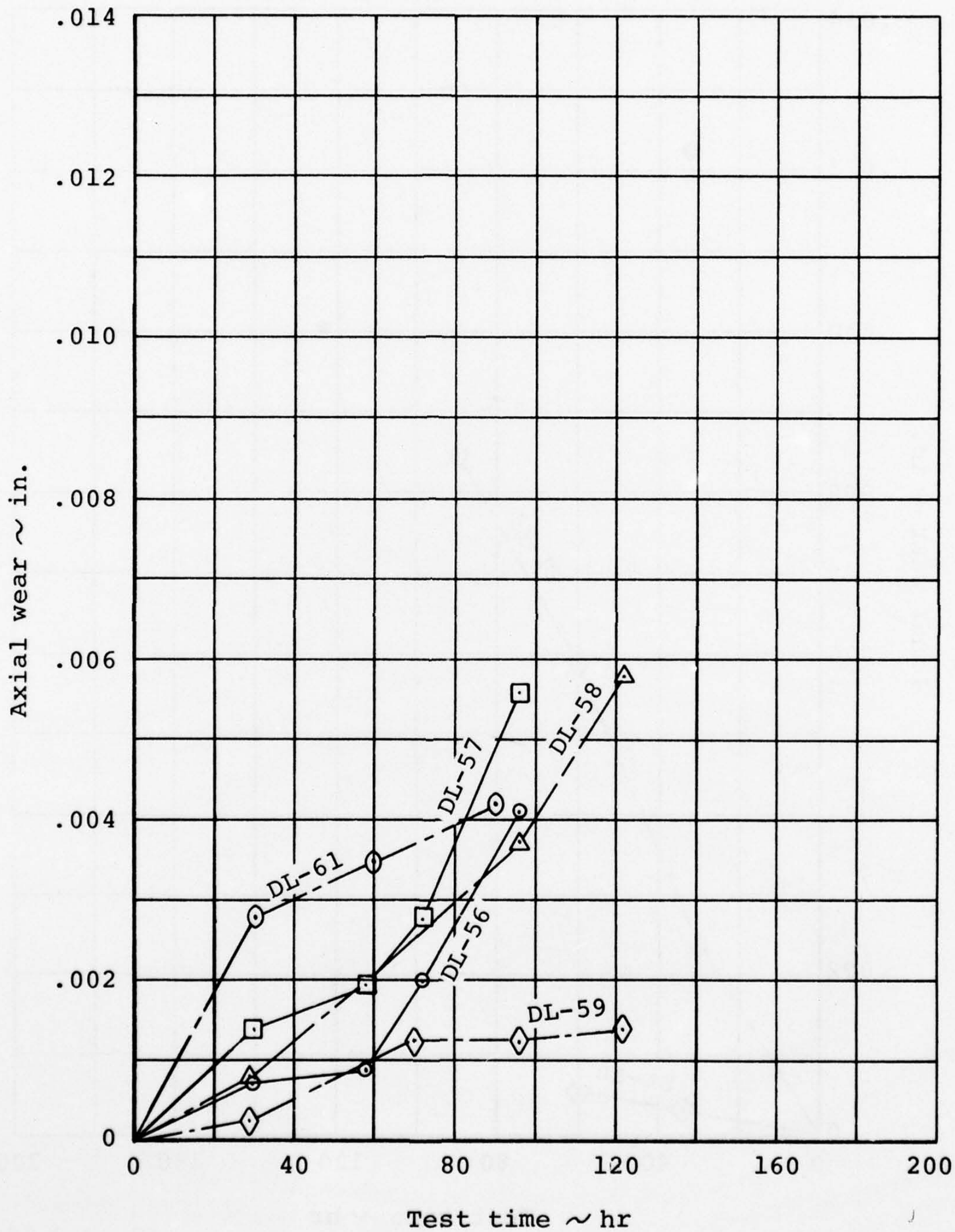


Figure 27. Axial wear vs test time - continuous rotation test rig.

TABLE 14. ANALYSIS OF VARIANCE FOR CONTINUOUS ROTATION SCREENING TESTS

SLOPES OF WEAR DATA (IN/HR x 100,000)

S.R.P. S(psi)	S.A.L. P (lb)	R=50 rpm(8.18fpm) A=6°	R=50 rpm(8.18fpm) A=0°	R=75 rpm(12.27 fpm) A=6°	R=75 rpm(12.27 fpm) A=0°
-4000	60	DL-58 12.21830			DL-56 6.61086
-4000	0		DL-4 7.30109	DL-30 2.72000	
-2000	60		DL-59 0.41233	DL-57 5.86177	
-2000	0	DL-2 0.43136			DL-31 1.77000

S.R.P. means static radial pressure, S.A.L. means static axial load, R means rotational speed, A means ball angle.

ANOVA TABLE

SOURCE OF VARIATION	SUM OF SQUARES	D.F.	MEAN SQUARES	F	Fcv	COMMENT
S & PRA	51.89151	1	51.89151	3.89	10.1	NSD
P & SRA	20.73941	1	20.73941	1.55	10.1	NSD
R & SPA	1.44538	1	1.44538	0.11	10.1	NSD
A & SPR	3.29879	1	3.29879	0.25	10.1	NSD
SP & RA	2.80291	1	2.80291			
SR & PA	36.02566	1	36.02566			
SA & PR	1.18922	1	1.18922			
	40.01779P	3P	13.33926P			
Total	117.39287	7				

F = mean squares ÷ 13.33926

$F_{.05(1,3)} = 10.1 = Fcv$

ANOVA TABLE (AFTER POOLING)

SOURCE OF VARIATION	SUM OF SQUARES	D.F.	MEAN SQUARES	F	Fcv	COMMENT
S & PRA	51.89151	1	51.89151	28.6	10.1	SIG
P & SRA	20.73941	1	20.73941	11.4	10.1	SIG
A & SPR	3.29879	1	3.29879	1.82	10.1	NSD
SR & PA	36.02566	1	36.02566	19.9	10.1	SIG
Residual	5.43751	3	1.81250			
Total	117.39287	7				

SIG Means there is a Significant Difference
NSD Means No Significant difference
P Means "Pooled"

out-of-plane angle and static radial pressure X surface velocity X ball out-of-plane angle, respectively. The third source of variation was the first-order interaction of static radial pressure X surface velocity which is confounded with the first-order interaction of static axial load X ball out-of-plane angle. Operating under the reasonably safe assumption that the two aforementioned main effects were significant as opposed to the second-order interactions with which they were confounded, the two validation test conditions were chosen to unconfound the two aforementioned first order interactions. Thus, test number 57 had the lowest static radial pressure x fpm but the highest static axial load x angle. Also, test number 58 was chosen to have the highest static radial pressure x fpm and the lowest static axial load x angle. Table 15 lists the test conditions for the validation tests.

TABLE 15. TEST CONDITIONS FOR CONTINUOUS ROTATION VALIDATION TESTS

TEST NO.	BALL ROTATIONAL SPEED (rpm)	STATIC RADIAL PRESSURE (psi)	STATIC AXIAL LOAD (lb)	BALL ANGLE (deg)	BALL SURFACE VELOCITY (fpm)	PV (psi-fpm)
57	50	2000C	60	6	8.18	16,362
58	75	4000C	0	0	12.27	49,086

STEPWISE MULTIPLE REGRESSION (CONTINUOUS ROTATION TEST RIG)

The fact that test number 57 caused low wear and test number 58 caused high wear pinpoints the static radial pressure x fpm interaction as being the significant interaction of the two previously mentioned confounded first-order interactions.

A stepwise multiple regression was performed with the data from the 8 screening tests and 2 validation tests. The results are presented in Table 16. These results are used in the Equation Development section of this report.

TABLE 16. RESULTS OF STEPWISE MULTIPLE REGRESSION (CONTINUOUS ROTATION TEST RIG)

Step 4

Variable Entered.....4 Dependent Variable.....5

Sum of Squares Reduced in this Step.........	0.043
Proportion Reduced in this Step.............	0.352
Cumulative Sum of Squares Reduced...........	0.097 of 0.122
Cumulative Proportion Reduced...............	0.798

For 4 Variables Entered

Multiple Correlation Coefficient...	0.893
(Adjusted for D.F.)............	0.835
F-Value for Analysis of Variance...	4.927
Standard Error of Estimate.........	0.070
(Adjusted for D.F.)............	0.086

VARIABLE NUMBER	REGRESSION COEFFICIENT	STD. ERROR OF REG. COEFF.	COMPUTED T-VALUE
1	0.42776	0.11570	3.697
2	0.00120	0.00077	1.564
3	-0.03534	0.01109	-3.187
4	0.10359	0.03512	2.950
Intercept	-1.13999		

Where: 1 = Static radial pressure ÷ 1000, psi
2 = Static axial load, lb
3 = Static radial pressure (surface velocity) ÷ 1000, psi-fpm
4 = Surface velocity, fpm

5 = Wear factor, $\frac{\text{in/hr}}{\text{psi-fpm}} \times 10^8$

Thus, Var. 5 = 0.42776(Var. 1)+0.00120(Var. 2)-0.03534(Var. 3) +0.10359(Var. 4)-1.13999 $\pm$0.140

To obtain k_i (Wear factor in in^3/lb-ft), divide the above equation by 60 which is the conversion factor from minutes to hours. The resultant equation appears on page 108 of this report.

WEAR TEST RESULTS (FOUR-BAY TEST RIG)

Two bearings were tested in the four-bay test rig. The results for bearings S/N 363 and 364 are presented in Table 17 and Figure 28. Table 17 lists the dates, test hours, load levels, and wear readings for these two bearings. The first 13 hours at 600-pound load was considered the break-in wear period for these bearings. Figure 28 is a plot of liner wear corrected for break-in wear versus test time for the two bearings.

TABLE 17. TABULATION OF WEAR DATA FOR #363 and #364 TEST BEARINGS

			363			364		
DATE	TEST HOURS	CUMULATIVE TEST HOURS	BAY NO.	LOAD (LB)	CUMULATIVE WEAR (IN.)	BAY NO.	LOAD (LB)	CUMULATIVE WEAR (IN.)
9/16/76	12.9	12.9	1	600	.0018	2	600	.0014
9/21/76	28.7	41.6	1	720	.0023	2	480	.0025
9/22/76	22.0	63.6	1	480	.0023	2	720	.0031
9/25/76	28.3	91.9	2	240	.0044	1	960	.0044
9/26/76	23.9	115.8	2	960	.0065	1	240	.0047
9/28/76	22.8	138.6	2	480	.0075	1	720	.0053
9/29/76	22.9	161.5	2	720	.0076	1	480	.0057
9/30/76	16.0	177.5	1	960	.0080	2	240	.0058
10/1/76	22.4	199.9	1	240	.0081	2	960	.0068
10/3/76	49.6	249.5	1	600	.0089	2	600	.0078
10/6/76	34.3	283.8	2	600	.0100	1	600	.0086
10/21/76	20.9	304.7	2	480	.0103	1	720	.0089
10/21/76	6.9	311.6	2	720	.0107	1	480	.0090
10/22/76	15.5	327.1	2	600	.0107	1	600	.0090
10/22/76	4.8	331.9	1	720	.0113	2	480	.0098
10/23/76	20.0	351.9	1	480	.0114	2	720	.0103
10/26/76	22.7	374.6	1	600	.0120	2	600	.0109
10/27/76	23.4	398.0	1	720	.0121	2	480	.0116
10/28/76	22.6	420.6	1	240	.0123	2	960	.0122
10/29/76	22.7	443.3	1	960	.0128	2	240	.0127
11/1/76	25.0	468.3	2	240	.0133	1	960	.0135
11/2/76	18.5	486.8	2	960	.0143	1	240	.0135
11/3/76	24.5	511.3	2	720	.0146	1	480	.0139
11/5/76	36.2	547.5	2	960	.0148	1	240	.0143
11/8/76	29.4	576.9	2	240	.0152	1	960	.0150
11/9/76	12.9	589.8	2	600	.0152	1	600	.0154
11/11/76	23.6	613.4	1	600	.0156	2	600	.0166
11/12/76	25.1	638.5	1	960	.0162	2	240	.0167

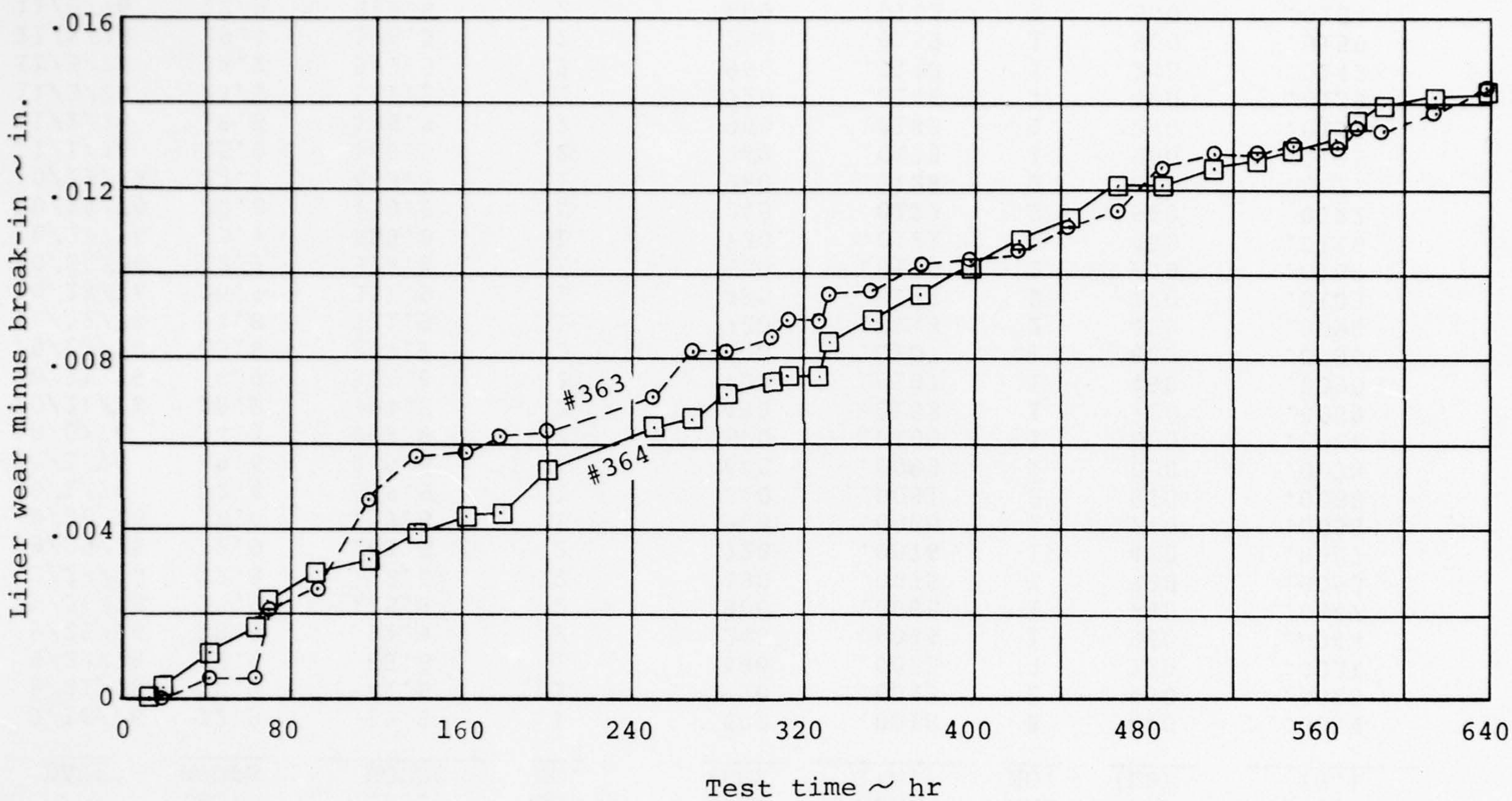


Figure 28. Liner wear minus break-in wear vs test time.

EQUATION DEVELOPMENT

EQUATION DEVELOPMENT (EMPIRICAL APPROACH)

Of the three analytical techniques for equation development utilized in this report, the most emphasis was placed on this technique. The computer program REGRESS, described in Appendix B, was used almost exclusively to develop the equations. Initial computer runs were made with only 36 observations to become familiar with the program, but as more test data became available it was commonplace to see computer runs with 284 or even 347 observations. For example, a computer run with 284 observations (almost all the wear measurements from the 36 screening tests on the twelve-bay test rig) was used to determine a preliminary equation for the total bearing wear. This equation, LUBE 12F-2-20, was used to predict the total wear for all levels of the primary variables for 350 hours of test time. As discussed in this report under the heading Selection of Test Conditions for Validation Tests, and also in Appendix C, this amounted to approximately 10,000 predictions, or wear calculations. From these 10,000 predicted values, twelve validation tests were chosen. Seven of these tests represented conditions producing low predicted values of wear and the other five represented high predicted values of wear.

Table 18 presents a comparison of the actual wear values versus the predicted values for the twelve validation tests. The column entitled Residual/S_Y was used to determine that eight of the twelve bearings had a predicted value which differed from the actual value by less than three times the standard error of the estimate. Table 18 also shows that equation LUBE 12F-2-20 correctly predicted the five bearings which would have high wear and the seven bearings which would have low wear.

However, equation LUBE 12F-2-20 with its twenty regression coefficients is obviously unwieldy and cumbersome to use. As will be noted from the large values of Residual/Sy in Table 18, there was need for improvement of the accuracy of the equation. The validation test data is real data and can be combined with the original data to derive a better predictive equation. The balance of the test conditions is usually sufficiently well preserved because the confirming tests constitute a small portion of the total. The result of combining the two sets of data is an improved equation and this equation could be confirmed by additional validation tests.

However, since all testing had been completed, a solution to this problem was evolved from the realization that all twelve of the validation tests were not needed for equation confirmation. In the consultant's previous experience, the number of

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DESIGN CRITERIA FOR DRY LUBRICATED FLIGHT CONTROL BEARINGS.(U)

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TABLE 18. COMPARISON OF ACTUAL AND PREDICTED WEAR VALUES FOR PRELIMINARY WEAR EQUATION LUBE 12F-2-20

CASE NO.	SPECIMEN NO.	BAY NO.	WEAR @ 350 HOURS (INCHES x 10^5) ACTUAL[a]	PREDICTED[b]	RESIDUAL	RESIDUAL/S_Y[d]
*1	37	1	150	172	-22	-0.27
35	38	2	155	-43	198	2.46
9a	39	3	105	336	-231	-2.87
39	40	4	25	-145	170	2.11
4c	41	5	1880	2222	-342	-4.25
3a	42	6	2115	2268	-153	-1.90
26	43	7	130	177	-47	-0.58
27	44	8	80	154	-74	-0.92
2a	45	9	(1420)[c]	(1056)[c]	(364)[c]	4.52
*2	46	10	60	202	-142	-1.76
1b	47	11	(1815)[c]	(1290)[c]	(525)[c]	6.52
2d	48	12	(1355)[c]	(1722)[c]	(-367)[c]	-4.56

Notes:

a. These values were obtained from Run number 4 of Table 9.

b. These values were predicted by equation LUBE 12F-2-20 and are listed in Table C-2.

c. The actual, predicted, and residual values listed in parentheses for specimen numbers 45, 47, and 48 are for 137, 175, and 250 hours, respectively.

d. The standard error of the estimate (S_Y) for LUBE 12F-2-20 is 80.5.

validation tests to be performed consisted of approximately ten percent of the number of tests used in equation development. Thus, for our case of 36 tests used for equation development, only four additional tests would normally be required for validation purposes. However, twelve tests were performed because twelve test bays were available.

In this contract a second predictive equation was obtained by combining data from eight of the validation tests with the data from the original 36 tests. The remaining four validation tests satisfied the 10 percent requirement for confirmation of the second equation.

By trial and error it was determined that the use of the test data from test specimens 39, 40, 45, and 47 for the final confirming tests not only provides a good balance for the confirming tests but also provides good balance for the remaining 44 test specimens. Table 19 presents a summary of the test balance for the original 36 screening tests, the 8 validation tests combined with the 36 screening tests, and the remaining 4 confirming tests. Table 19 also shows that the final confirming tests consisted of two bearings with low predicted wear and two bearings with high predicted wear.

Using the test data from Table 9 for the 44 test specimens, a computer run with 338 observations provided a new predictive equation for total liner wear, LUBE 23H-1-7. Nine terminal wear values were not included in the regression analysis because nonlinear wear had occurred, indicating catastrophic wear had been encountered. See Figures 10, 11, 12, 14, 15, 16, 17, and 20. As shown in Figure 29, this equation had only seven regression coefficients in its most general form because the next most important term would have explained only 0.2% of the wear variation. It would complicate the wear equation with no substantial increase in accuracy. (See Appendix G for a list of the 72 variables utilized in the regression and from which the seven most significant variables were selected.)

The equation was further reduced to only four regression coefficients by making two equations out of the one basic equation (one equation to be used for bearings with water contamination and the other equation for bearings with no water contamination). The reduction in the number of regression coefficients resulted from substituting a 2 for variable X_5 for the water contamination case and a 1 for X_5 if water was not present. Thus, the regression coefficient associated with variable X_5 disappears and results in a change to the intercept, 315.33. Similarly, the regression coefficients for the water-time and water-cyclic radial pressure interactions disappear and result

TABLE 19. SUMMARY OF TEST BALANCE

CODE	VARIABLE	LEVEL	36 TESTS	44 TESTS	4 TESTS
			NUMBER OF SPECIMENS		
A	Static Radial Pressure (psi)	2000C	9	11	1
		1000C	9	10	0
		0	9	11	1
		1000T	0	1	0
		2000T	9	11	2
B	Speed of Ball Oscillation (cpm)	300	12	14	2
		600	12	16	0
		900	12	14	2
C	Ball Oscillating Angle (deg)	5	12	14	3
		10	12	16	0
		15	12	14	1
D	Cyclic Radial Pressure (psi)	0	9	10	1
		1000	9	11	0
		1500	9	11	0
		2000	9	12	3
E	Phase Angle Between C&D (deg)	0	12	17	3
		45	12	14	0
		90	12	13	1
F	Static Axial Load (lb)	0	12	16	2
		30	12	14	0
		60	12	14	2
G	Contaminants	None	8	9	0
		S&D	7	8	0
		680	7	8	1
		500A	7	9	1
		Water	7	10	2
-	Predicted Wear	Low	--	--	2
		High	--	--	2

TOTAL LINER WEAR

$$W = 315.33 + 4.67353(X_5)(X_{11}) - 4.48547(X_{11}) - 295.36(X_5) + 108.38(X_{28}) \times 10^{-6} + 9.9933(X_6)^2 \times 10^{-6} + 0.17812(X_5)(X_9) - 0.18567(X_9) \pm 220.9$$

LINER WEAR ON COMPRESSION SIDE

$$W_c = 272.32 + 3.6924(X_5)(X_{11}) - 3.5310(X_{11}) - 0.06230(X_5)(X_6) - 265.87(X_5) + 0.08888(X_6) - 0.0056421(X_5)(X_6)(X_8) \pm 186.8$$

LINER WEAR ON TENSION SIDE

$$W_T = 127.20 + 3.8139(X_5)(X_{11}) + 0.33333(X_5)(X_6) - 128.98(X_5) - 0.30576(X_6) - 3.7014(X_{11}) + 5.2783(X_{13}) - 5.2817(X_5)(X_{13}) \pm 184.0$$

RADIAL PLAY

$$R.P. = +208.58 + 3.8955(X_5)(X_{11}) - 3.8045(X_{11}) - 184.47(X_5) - 16.673(X_5)(X_{10}) + 16.254(X_{10}) - 0.87493(X_8)(X_{10}) - 0.19939(X_9) + 0.19987(X_5)(X_9) + 914.78(X_5)(X_8)(X_{10}) \times 10^{-3} \pm 167.6$$

DEFINITION OF VARIABLES

- X_5 = water (use $X_5 = 2$ if water is present; otherwise, $X_5 = 1$)
- X_6 = static radial pressure, psi (+ = tension in rod end shank)
- X_8 = angle of ball oscillation, deg (where one cycle = $4(X_8)$)
- X_9 = cyclic radial pressure, psi (total radial pressure = $X_6 \pm X_9$)
- X_{10} = static axial load, lb
- X_{11} = time, hr
- X_{13} = phase angle between cyclic radial pressure and ball oscillation angle, deg
- X_{25} = cyclic radial pressure x 1, 0.707, or 0.5 for X_{13} = 0°, 45°, or 90° respectively
- X_{27} = cyclic work, psi-fpm; $X_{27} = (X_{25})(V)$ (use same sign as X_6)
- X_{28} = absolute value of PV, psi-fpm

Figure 29. General form of wear equations.

TABLE 20. COMPARISON OF ACTUAL AND PREDICTED WEAR VALUES FOR FINAL WEAR EQUATION LUBE 23H-1-7

	CASE NO.	SPECIMEN NO.	BAY NO.	WEAR @ 350 HOURS (INCHES x 10^5) ACTUAL[a]	PREDICTED[b]	RESIDUAL	RESIDUAL/Sy[d]
	*1	37	1	150	127	23	0.21
	35	38	2	155	113	42	0.38
	4c	41	5	(1115)[c]	(1122)[c]	(-7)[c]	-0.06
	3a	42	6	(1475)[c]	(1293)[c]	(182)[c]	1.65
	26	43	7	130	121	9	0.08
	27	44	8	80	76	4	0.04
	*2	46	10	60	77	-17	-0.15
	2d	48	12	(990)[c]	(967)[c]	(-23)[c]	-0.21
Confirming tests	9a	39	3	105	126	-21	-0.19
Confirming tests	39	40	4	25	112	-87	-0.79
Confirming tests	2a	45	9	(1030)[c]	(595)[c]	(435)[c]	3.94
Confirming tests	1b	47	11	(1085)[c]	(557)[c]	(528)[c]	4.78

Notes:

a. These values were obtained from Run number 4 of Table 9.

b. These values were predicted by equation LUBE 23H-1-7.

c. The actual, predicted, and residual values listed in parentheses for specimen numbers 41, 42, 45, 47, and 48 are for 250, 250, 100, 100, and 175 hours, respectively.

d. The standard error of the estimate (Sy) for LUBE 23H-1-7 is 110.5.

in changes to the coefficients for the time and cyclic radial pressure variables, respectively.

Table 20 presents a comparison of the actual and predicted wear values for the twelve bearings tested in run number 4. The Residual/Sy column shows that two out of the four confirming tests have values greater than ± 3 Sy. These are the two bearings which were contaminated with water. Values of ± 3 standard errors of the estimate are conventionally used confidence limits for verifying the ability of regression equations to predict actual outcomes. In a Gaussian distribution, 99.7% of all the values will fall within these limits. When actual results come outside of these limits, it indicates a most unusual occurrence or that the equation does not predict well. Therefore, it is concluded that equation LUBE 23H-1-7 may not predict wear life within ± 3 times the standard error of the estimate for the very short lives that result when water is present.

Equations for wear on the compression side of the bearing, wear on the tension side, and radial play were also developed by use of the stepwise multiple regression process. Appendix G lists the 72 independent variables utilized in computer program REGRESS and Appendix H describes the equation development. Each of the predictive equations is shown in Figure 29 and was resolved into two equations (one equation for water contamination and the other for no water contamination) in a similar manner as done for the equations for total liner wear. Table 21 presents a summary of the correlation coefficients, the coefficients of variation, and the standard errors of the estimate for the four general equations.

TABLE 21. STATISTICS OF FINAL PREDICTIVE EQUATIONS

VARIABLE	EQUATION	r	r^2	Sy	Cv (%)	F-value	$\bar{x}$	σx	Sy/σx	k	n-k-1	Fcr
W	23H-1-7	.900	.810	110.5	76.9	200.4	143.6	250.4	.441	7	330	3.08
W_C	24A-1-6	.896	.804	93.4	79.3	162.2	117.8	208.1	.449	6	238	3.28
W_T	24G-1-7	.898	.807	92.0	116.5	117.1	78.95	205.9	.447	7	196	3.08
R.P.	25A-1-9	.934	.872	83.8	73.3	195.5	114.4	230.1	.364	9	259	2.80

Notes:

1. r = Correlation coefficient.
2. r^2 = Coefficient of determination which is the ratio of the explained variation in the dependent variable to the total variation in the dependent variable.
3. Sy = Standard error of the estimate.
4. C_v = Coefficient of variation which is the ratio of Sy to $\bar{x}$ in percent.
5. F-value = The computed value from Analysis of Variance performed by the Stepwise Multiple Regression computer program.
6. $\bar{x}$ = Mean of the n observations of the dependent variable.
7. σ_x = Standard deviation of the n observations of the dependent variable.
8. Sy/σ_x = Ratio of standard error of estimate to standard deviation.
9. k = Number of independent variables used in the regression.
10. n = Number of observations used for the regression.
11. F_{cr} = Critical value for the F-value based on k degrees of freedom in the numerator, n-k-1 degrees of freedom in the denominator, and .005 level of significance.
12. F-value must be larger than F_{cr} in order for the linear regression of the dependent variable on the k independent variables jointly to be significant.

EQUATION DEVELOPMENT (DETERMINISTIC APPROACH)

Initial attempts at modification of existing equations to fit the twelve-bay test rig data centered on the equation given in nomograph form by Rexnord, Inc., Downers Grove, Illinois. See Appendix D for an explanation of the unsuccessful attempts with the Rexnord equation.

Better results were obtained from an equation which has been reported in many Teflon bearing articles and catalogs. One useful form of the equation equates the ratio of wear rate and PV to the wear factor, k, as follows:

$$k = \frac{W.R.}{PV} \text{ where}$$

W.R. = wear rate, in/min
P = pressure, psi
V = velocity, fpm
k = wear factor, $in^3/lb\text{-}ft$ (values for k are listed in bearing manufacturer's catalogs and are normally for low-speed, fixed-wing applications.)

In our work with the above equation we have used the following parameters:

$$P = |S.R.P.| + c|C.R.P.|$$

S.R.P. = static radial pressure, psi
C.R.P. = cyclic radial pressure, psi
c = 1, 0.707, or 0.5 if X_{13} = 0°, 45°, or 90°, respectively
X_{13} = phase angle between cyclic radial pressure and ball oscillation angle, deg

$$V = \frac{\pi}{1080}(B.D.)(X_7)(X_8)$$

B.D. = ball diameter, in
X_7 = speed of ball oscillation, cpm
X_8 = angle of ball oscillation, deg, where one cycle = $4(X_8)$
W.R. = slope of wear curves presented in Table 12 (page 82) divided by 60
T = time, hr

Using the W.R./PV equation, we obtained 48 values of wear factor for the 48 bearings tested in the twelve-bay test rig. The average wear factor for the 36 bearings tested without water calculated to be 0.172×10^{-11} $in^3/lb\text{-}ft$ with a standard deviation of 0.244×10^{-11} $in^3/lb\text{-}ft$. The average wear factor for the 12 bearings tested with water calculated to be 7.239×10^{-11} $in^3/lb\text{-}ft$ with a standard deviation of 6.999×10^{-11} $in^3/lb\text{-}ft$. These standard deviations represent considerable variability in the wear factor. Bearing manufacturers normally list one constant value for wear factor without indicating the variability which the user can expect.

In order to approximate the variation in wear factor, two stepwise multiple regression runs were done using the 36 non-water wear factor values for the first run and the 12 water wear factors for the second run. Thus, two equations for wear factor were obtained.

Without water

$$k = [377.36 - 0.19676(\text{C.R.P.}) + 0.49785(\phi)] \times 10^{-14} \pm 411.4 \times 10^{-14}$$

With water

$$k = [18.117 - 0.00721(\text{C.R.P.}) - 0.03333(\phi) + 0.00077(\text{S.R.P.})] \times 10^{-11} \pm 9.968 \times 10^{-11}$$

The statistics of the two regression analyses are listed below:

36 bearings without water contamination		12 bearings with water contamination	
$r = 0.573$	$r^2 = 0.328$	$r = 0.795$	$r^2 = 0.632$
$S_y = 205.7 \times 10^{-14}$	$S_{y/\sigma} = 0.845$	$S_y = 4.984 \times 10^{-11}$	$S_{y/\sigma} = 0.712$
$C_v = 119.6\%$		$C_v = 68.8\%$	

Thus, the equation, Wear = 60k PVT, can be used to approximate the wear life of the bearings used in this program by simply calculating the particular wear factor for water or no water. Based on the values of r^2 listed above, the equations for k explain only 32.8% and 63.2% of the variation in wear factor obtained for the bearings tested without water and with water, respectively.

EQUATION DEVELOPMENT (THEORETICAL APPROACH, COROLLARY NUMBER 1)

As explained in the Techniques for Equation Development section of this report, two test bearings were tested 638.5 hours for proof of corollary number 1. The results are presented in the Test Results section of this report and were used to calculate wear rates. A three-way analysis of variance was performed utilizing 40 wear rates, 20 for each bearing, with two replications. The results showed that, at the .05 level of significance, the wear rate was affected by the load levels and the particular test bay being used. Table 22 presents the ANOVA tables. There was no significant difference in wear rate caused by the replication performed after the first 284 hours of testing. Most importantly, the analysis of variance showed that there was no significant difference between the two test bearings.

This testing was encouraging and helps to lend credence to the corollary that the wear of a bearing is not affected by the duty cycle within the limits tested. More testing with varying speeds and angles would be required to add further proof.

EQUATION DEVELOPMENT (THEORETICAL APPROACH, COROLLARY NUMBER 2)

The development of an equation based on the incremental wear theory was started by defining a general sinusoidal relationship for velocity and pressure as shown in Figure 30. The velocity and pressure sinusoids were purposely shown offset by the phase angle ϕ to keep the analysis general. The method now consisted of subdividing one cycle of the velocity sinusoid into n partitions. A similar number of partitions were used for the pressure sinusoid. Thus, for position θ_i in the velocity cycle, we calculated the $[PV]_i$ value to be as shown in Figure 30. For the partition $d\theta_i$ at position θ_i, we can calculate the increment of wear, wear $_{d\theta_i}$, by using equation, wear = 60k (PV)t. Thus, wear $_{d\theta_i} = 60k_i [PV]_i \, t_{d\theta_i}$ where the wear factor, k_i, was obtained from the continuous rotation test rig results and $t_{d\theta_i}$ was the time calculated in hours for the partition $d\theta_i$. The total wear for one cycle will equal the summation of the complete cycle. Thus, total wear = $\Sigma_{i=1}^{n}$ (wear $_{d\theta_i}$).

Then, wear for t hours can be calculated by multiplying the total wear for one cycle by 60(t) (CPM) where CPM is the speed of the ball oscillation and 60 is the conversion factor from minutes to hours.

TABLE 22. ANALYSIS OF VARIANCE OF WEAR RATES IN FOUR-BAY RIG

SOURCE OF VARIATION	SUM OF SQUARES	DEGREES OF FREEDOM	MEAN SQUARES	F	F_{CV}	COMMENT
A (Test Bays)	2877.87	1	2877.87	10.87	4.28	SIG
B (Loads)	3314.00	4	828.50	3.13	2.80	SIG
C (Bearings)	106.71	1	106.71	0.40	4.28	NSD
Replication	679.52	1	679.52	2.57	4.28	NSD
A*B	623.05	4	155.76	0.59	2.80	NSD
A*C	801.15	1	801.15	3.03	4.28	NSD
B*C	831.15	4	207.79	0.79	2.80	NSD
A*B*C	P 1304.69	P 4	P 326.17	1.30	2.90	NSD
Residual	4783.12	19	251.74			
(pooled)	6087.81	23	264.69			
Total	15321.30	39				

$F_{.05(4,\ 19)} = 2.9$ $F_{.05(1,\ 23)} = 4.28$

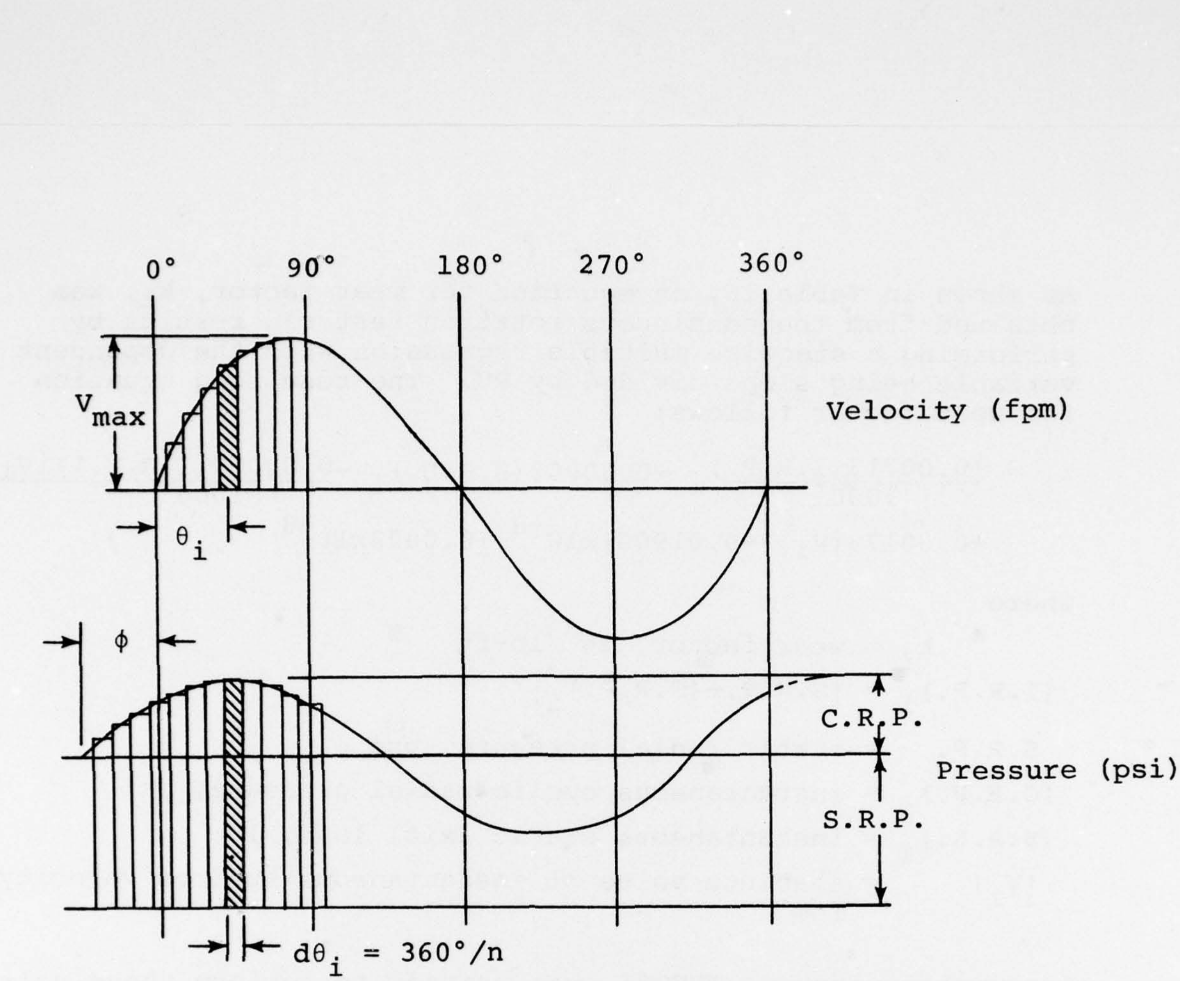
$F_{.05(4,\ 23)} = 2.80$

REVISED ANOVA TABLE AFTER SECOND POOLING

SOURCE OF VARIATION	SUM OF SQUARES	DEGREES OF FREEDOM	MEAN SQUARES	F	Fcv	COMMENT
A (Test Bays)	2877.87	1	2877.87	11.04	4.15	SIG
B (Loads)	3314.00	4	3314.00	3.18	2.67	SIG
C (Bearings)	106.71	1	106.71	0.41	4.15	NSD
Replication	679.52	1	679.52	2.61	4.15	NSD
Residual	8343.16	32	260.72			
Total	15321.30	39				

$F_{.05(1,\ 32)} = 4.15$ $F_{.05(4,\ 32)} = 2.67$

Where: SIG means there is a significant difference
NSD means no significant difference
P means "pooled"



At position θ_i, we have $V_i = V_{MAX}\sin\theta_i$, $t_{d\theta_i} = \dfrac{1}{60n(CPM)}$

$$P_i = S.R.P. + C.R.P.\ \sin\ (\theta_i + \phi)$$

$$[PV]_i = P_i(V_i) = (S.R.P.)V_{MAX}\sin\theta_i + (C.R.P.)V_{MAX}(\sin\theta_i)\sin(\theta_i + \phi)$$

where

V_i and P_i are the instantaneous values of velocity and pressure at θ_i

$$V_{MAX} = \pi(B.D.)\ \frac{4\theta}{360}\ \frac{(CPM)}{12} = \frac{\pi(B.D.)\theta(CPM)}{1080},\ \text{fpm}$$

B.D. = ball diameter, in
θ = angle of ball oscillation, deg (one cycle = 4θ)
CPM = speed of ball oscillation, cpm
θ_i = instantaneous angular position of ball oscillation motion cam, deg
S.R.P. = static radial pressure, psi
C.R.P. = cyclic radial pressure, psi
ϕ = phase angle, deg
$d\theta_i$ = size of subdivision of one cycle, deg
n = number of partitions of one cycle
$t_{d\theta_i}$ = time equivalent to $d\theta_i$, hr

Figure 30. Basic equations for theoretical approach to equation development.

As shown in Table 16, an equation for wear factor, k_i, was obtained from the continuous rotation test rig results by performing a stepwise multiple regression with the dependent variable being slope divided by PV. The resulting equation for wear factor follows:

$$k_i = [\frac{0.00713(I.R.P.)_i}{1000} + 0.00002(S.A.L.)_i - \frac{0.00059(I.R.P.)_i|V_i|}{1000}$$
$$+0.00173|V_i| - 0.01900]\times 10^{-8} \pm 0.0023\times 10^{-8}$$

where

k_i = wear factor, $in^3/lb\text{-}ft$

$(I.R.P.)_i$ = $|S.R.P. + (C.R.P.)_i|$

S.R.P. = static radial pressure, psi

$(C.R.P.)_i$ = instantaneous cyclic radial pressure, psi

$(S.A.L.)_i$ = instantaneous static axial load, lb

$|V_i|$ = absolute value of instantaneous surface velocity, fpm

A computer program, THEOR1, was written to perform these calculations. The program allowed the engineer to subdivide the velocity cycle into 576 increments, thus approaching an exact solution which would have been obtained if integral calculus had been used.

The calculated wear values obtained through use of this program are exceptionally high. The reason for the high values was traced to the high values of wear obtained in the continuous rotation test rig. Even though the test conditions had been chosen to provide static radial pressure and PV values consistent with the values used on the twelve-bay test rig, the test bearings in the continuous rotation test rig ran hot continuously whereas those in the twelve-bay test rig ran hot for only the first few hours. The wear debris exuding from the test bearings on the continuous rotation test rig was black, sticky, and in no way resembled the wear debris seen in the twelve-bay test rig. Obviously, too much energy was being introduced into the bearings in the continuous rotation test rig and consequently caused excessive and unrealistic wear. Thus, there is inconclusive proof of corollary No. 2.

WEAR LIFE EQUATIONS

INTRODUCTION

This section of the report presents the equations which can be used by the control designer to approximate the wear and radial play to be expected in control bearings. The equations can be used to predict wear and radial play from the end of the break-in wear regime to almost the end of the linear wear regime. The three separate and distinct wear regimes encountered in the population of the 12 bearings tested with water contamination are consistent with our previous experience with similar Teflon bearings. We chose to name these regimes the break-in wear regime, the linear wear regime, and the catastrophic wear regime. The high wear rates of the catastrophic wear regime occurred in almost all cases well above our selected safe limit of 0.012 inch liner thickness. The catastrophic wear regime was not encountered in the population of 36 bearings tested without water contamination since a maximum of only 0.002 to 0.003 inch liner wear was obtained in 350 hours of testing.

The equations presented in this section are linear because the population of 48 bearings demonstrated linear behavior. Although we do not claim an exhaustive search for nonlinearity, it is important to note that we tried the following nonlinear functions: The natural logarithm of time, the natural logarithm of wear, the natural logarithm of radial play, the square of static radial pressure, and the cube of static radial pressure. No better fit to the test data was found.

The wear life equations are a compromise between easily manageable equations and highly accurate equations. There is a point of diminishing returns where the addition of extra variables to improve accuracy conflicts with the need for ease of calculation.

The designer will be required to determine the conditions that his proposed or already existent vehicle will impose on the control bearings. Once these conditions are identified, he can obtain predictions of wear and radial play by use of the equations presented.

EQUATION LIMITATIONS

The wear life equations given herein have been derived by statistical process from the test data generated in this contract. These equations are valid for Type I spherical flight control bearings fabricated in general conformance with MIL-B-81819, Draft #5 and MS14101-6 Bearing Standards by Rexnord, with the Rexlon -2 liner system, and operating within the limits of the conditions tested. These equations may not be valid for other sizes, manufacturers, or operating conditions.

However, as discussed in Appendix E, other test data indicate bearing size has no significant effect between the -6 and -16 sizes. Thus, the wear life equations should be valid for all sizes between -6 and -16. Also, the best three out of eight manufacturers had no significant difference in wear life, indicating that equations found valid for one of the manufacturers would be valid for the other two. It is conjectured that bearings with liner systems proven equivalent to the Rexlon -2 liner system could be selected via the wear life equations with the same level of confidence.

The equations for bearings with water contamination must be used with caution because the predictions for the two confirming tests did not fall within ± 3 standard errors of the estimate. The equations for bearings without water contamination are based on testing of 350 hours and liner wear of 0.003 inches or less. The equations for bearings without water contamination result in considerable extrapolation for very long wear life predictions, which may cause concern. However, these equations are believed to be the best information available. They constitute a guide, to be used with care and improved with use.

The equations can be used to calculate wear for two conditions, with water and without water. The question of multiple contaminants arises, such as water plus sand and dust. Multiple contaminants were not tested in this program. However, it can be observed that the corrosion products resulting from water are abrasive and at least as destructive as sand and dust. Therefore, the addition of sand and dust to water contamination probably will result in a wear life equivalent to the water-contaminated wear life prediction. It is suggested that the wear-life predicted with water present can be used for the case where water and sand and dust are present.

LINER WEAR LIMIT

After the total wear equation has been used to determine the predicted wear for a particular set of conditions, the designer must use the equations for wear on the compression side and wear on the tension side to determine whether more than 0.012 inch of liner wear will occur on either side. It is recommended that liner wear of more than 0.012 inch on each side be avoided with the Rexlon -2 liner system because 0.012 inch liner wear was observed in the testing to be a safe limit.

It will be important for the designer to consider whether his application is sensitive to radial play or to wear or both. In many cases the designer will find that the proposed bearing will be unacceptable for further use because of intolerable radial play even though much of the bearing liner remains to be worn. Also, one bearing could conceivably run twice as long as another even though they both experience identical total liner wear because the total liner wear could be divided equally between the tension and compression side of the first bearing and the total liner wear could be concentrated all on the tension side (or compression side) of the second bearing.

TOTAL LINER WEAR EQUATIONS

Total liner wear can be calculated with Equation A for bearings not contaminated with water and Equation B for bearings with water.

EQUATION A (Bearings without water contamination)

TOTAL LINER WEAR

$$W = +19.97 + 9.9933(X_6)^2 \times 10^{-6} - 0.00755(X_9) + 0.18806(X_{11}) + 108.38(X_{28}) \times 10^{-6} \pm 220.9$$

EQUATION B (Bearings with water contamination)

TOTAL LINER WEAR

$$W = -275.4 + 9.9933(X_6)^2 \times 10^{-6} + 0.17057(X_9) + 4.86159(X_{11}) + 108.38(X_{28}) \times 10^{-6} \pm 220.9$$

ADDITIONAL WEAR EQUATIONS AND RADIAL PLAY EQUATIONS

Once the total liner wear has been calculated, either equations C for bearings with no water contamination or equations D for bearings with water should be used to check for wear on the compression (tension) side and also radial play, if required. Warning: Do not calculate W_C for a bearing with only tension loads. Do not calculate W_T for a bearing with only compression loads. Also, the radial play (R.P.) equation includes negative wear, i.e., redeposited wear debris, which affects radial play. R.P. will be less than W for a bearing with only a tension or a compression load.

EQUATIONS C (Bearings without water contamination)

$$W_C = 6.4 + 0.02658(X_6) - 0.0056421(X_6)(X_8) + 0.1614(X_{11}) \pm 186.8$$

$$W_T = -1.8 + 0.02757(X_6) + 0.11250(X_{11}) - 0.00340(X_{13}) \pm 184.0$$

$$R.P. = +24.1 - 0.4190(X_{10}) + 0.03985(X_8)(X_{10}) + 0.00048(X_9) + 0.0910(X_{11}) \pm 167.6$$

EQUATIONS D (Bearings with water contamination)

$$W_C = -259.4 - 0.03572(X_6) - 0.011284(X_6)(X_8) + 3.8538(X_{11}) \pm 186.8$$

$$W_T = -130.8 + 0.36090(X_6) + 3.9264(X_{11}) - 5.2851(X_{13}) \pm 184.0$$

$$R.P. = -160.4 - 17.092(X_{10}) + 0.95463(X_8)(X_{10}) + 0.20035(X_9) + 3.9865(X_{11}) \pm 167.6$$

DEFINITION OF VARIABLES

X_6 = static radial pressure, psi (+ = tension in rod end shank)

X_7 = speed of ball oscillation, cpm

X_8 = angle of ball oscillation, deg, where one cycle = $4(X_8)$

X_9 = cyclic radial pressure, psi (total radial pressure = $X_6 \pm X_9$)

X_{10} = static axial load, lb

X_{11} = time, hr

X_{13} = phase angle between cyclic radial pressure and ball oscillation angle, deg

X_{25} = cyclic radial pressure x 1, 0.707, or 0.5 for X_{13} = 0°, 45°, or 90°, respectively

X_{27} = cyclic work, psi-fpm; $X_{27} = (X_{25})(V)$ (use same sign as X_6)

X_{28} = absolute value of PV, psi-fpm

W = total liner wear, in x 100,000

W_c = liner wear on compression side, in x 100,000 (+ means wear, - means buildup or wear debris)

W_T = liner wear on tension side, in x 100,000 (+ means wear, - means buildup of wear debris)

R.P.= radial play, in x 100,000

PV = $(X_6)V + (X_{25})V$ where X_{25} is + when X_6 is +; X_{25} is - when X_6 is -.

V = $\pi \frac{B.D.}{12}(X_7)\frac{4(X_8)}{360}$, fpm

B.D.= ball diameter, in

NOMOGRAPHS FOR GRAPHICAL SOLUTION OF REQUIRED VARIABLES

Figure 31 is a nomograph which can be used for rapid estimation of the surface velocity (V) in fpm. The use of the nomograph is self-explanatory. Figure 32 is a nomograph which allows rapid estimation of variables X_{25}, X_{27}, and X_{28}. Variable X_{25} can be obtained from Figure 32 by:

1. drawing a horizontal line from the value of cyclic radial pressure (1000) to intersect with the diagonal line for the phase angle (90°).

2. drawing a vertical line from the aforementioned intersection to the cyclic radial pressure modified (X_{25}) scale and interpolating the answer (500).

Variable X_{27} can be obtained by:

3. locating the value for surface velocity on the vertical surface velocity scale (16.36).

4. drawing a straight line from the surface velocity scale to the lower left zero reference point.

5. drawing a horizontal line from the intersection of the vertical line drawn in 2 above with the diagonal line drawn in 4 above and interpolating the answer (8200).

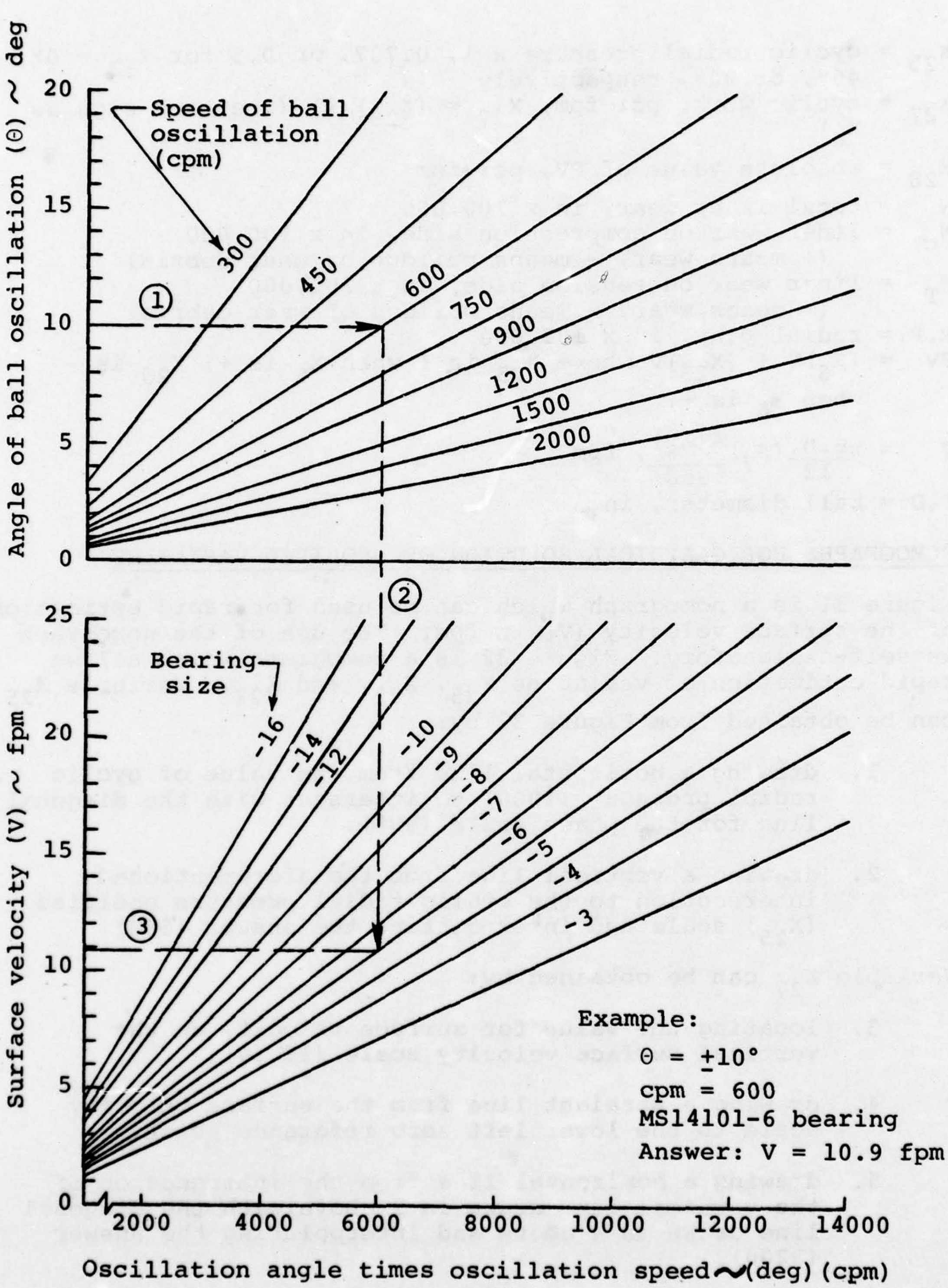


Figure 31. Surface velocity nomograph.

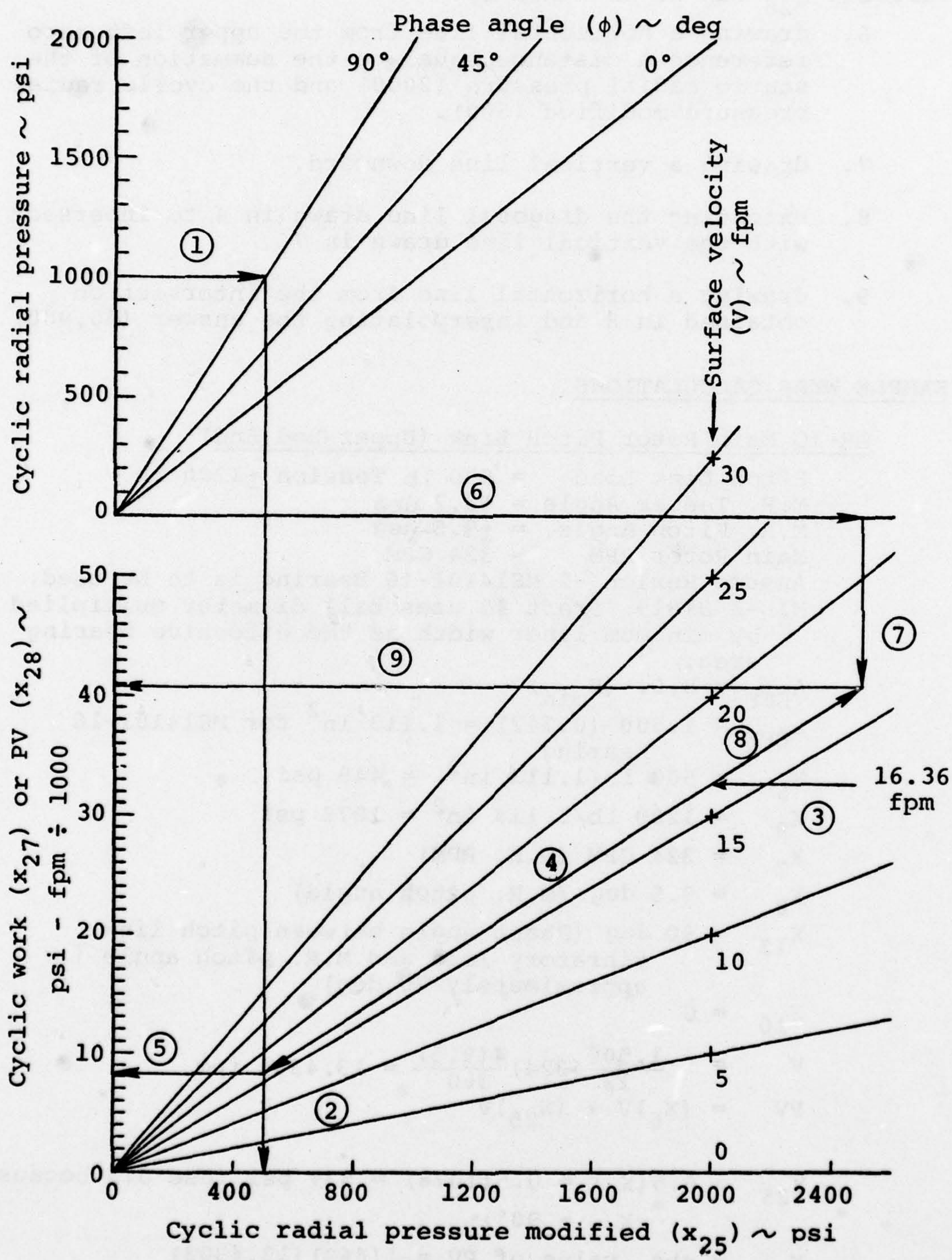


Figure 32. Nomograph for x_{25}, x_{27}, and x_{28}.

Variable X_{28} can be obtained by:

6. drawing a horizontal line from the upper left zero reference a distance equal to the summation of the static radial pressure (2000) and the cyclic radial pressure modified (500).

7. drawing a vertical line downward.

8. extending the diagonal line drawn in 4 to intersect with the vertical line drawn in 7.

9. drawing a horizontal line from the intersection obtained in 8 and interpolating the answer (40,900).

SAMPLE WEAR CALCULATIONS

AH-1Q Main Rotor Pitch Link (Upper Rod End)

Pitch Link Load = 500 lb Tension $\pm$1200 lb
M.R. Teeter Angle = $\pm$3.2 deg
M.R. Pitch Angle = $\pm$9.5 deg
Main Rotor RPM = 324 CPM
Assume Rexlon -2 MS14101-16 Bearing is to be used.
MIL-B-81819, Draft #5 uses ball diameter multiplied by minimum liner width as the effective bearing area.

A_{EFF} = B.D. (H_{min})

A_{EFF} = 1.500 (0.742) = 1.113 in^2 for MS14101-16 bearing

X_6 = 500 lb/1.113 in^2 = 449 psi

X_9 = 1200 lb/1.113 in^2 = 1078 psi

X_7 = 324 CPM (M.R. RPM)

X_8 = 9.5 deg (M.R. pitch angle)

X_{13} = 90 deg (Phase angle between pitch link vibratory load and M.R. pitch angle is approximately 90 deg)

X_{10} = 0

$$V = \pi \frac{1.500}{12}(324)\frac{4(9.5)}{360} = 13.4303 \text{ fpm}$$

$$PV = (X_6)V + (X_{25})V$$

X_{25} = 0.5(X_9) = 0.5(1078) = 539 psi (Use 0.5 because X_{13} = 90°)

X_{28} = abs. value of PV = |(449)(13.4303) +(539)(13.4303)|
= 13269 psi-fpm

Assume Water Contamination. Assume the maximum total liner wear for this application has been specified to be 0.020 inch and the desired wear life is 1000 hours. Therefore, X_{11} = 1000 hr.

Use equation B, page 111 and the W_T equation from equations D, page 112 to calculate total liner wear and liner wear on the tension side.

(It is recommended that liner wear on the compression side not be calculated because the loading is predominantly tension. Also, radial play is not calculated because both tension and compression loads are present.)

$$W = -275.4 + 2.015 + 183.87 + 4861.59 + 1.438 = 4773.5$$

$$W_T = -130.8 + 162.04 + 3926.4 - 475.66 = 3482.0$$

Thus, $W = 0.0477 \pm 0.0022$ in (which exceeds 0.020 in maximum)

$W_T = 0.0348 \pm 0.0018$ in (which exceeds 0.012 in liner wear limit)

Using $W = 2000$ and $W_T = 1200$, solve above equations for time, X_{11}.

$$W = 2000 = -275.4 + 2.015 + 183.87 + 4.86159(X_{11})_W + 1.438 \pm 220.9$$

$$(X_{11})_W = 430 \text{ hrs} \pm 45 \text{ hrs}$$

$$W_T = 1200 = -130.8 + 162.04 + 3.9264\ (X_{11})_{WT} - 475.66 \pm 184.0$$

$$(X_{11})_{W_T} = 419 \text{ hrs} \pm 47 \text{ hrs}$$

The calculated minimum life with water present is 419-47 = 372 hrs.

CONCLUSIONS

1. Two separate and distinct populations were identified from the 48 bearings tested in the twelve-bay test rig. One population consisted of 36 bearings with no water contamination and the second population consisted of 12 bearings with water contamination.

2. The average wear characteristics of the 12 bearings with water contamination were identical to the average for the 36 bearings without water for the first 25 hours of test time. After this break-in period, the water contaminated bearings exhibited an average wear rate approximately 60 times as great as the average wear rate for the bearings not contaminated with water.

3. All the twelve-bay wear data were linear with time unless more than 0.012 inch liner wear was encountered.

4. The major contributors to total liner wear are water interacting with time, water interacting with cyclic radial pressure, static radial pressure, and maximum absolute value of PV. These four contributors explain 81% of the variation in total liner wear.

5. The wear predictions are poor (residual/Sy > $\pm$ 3 Sy) for the very short lives that result when water is present.

6. Load ratio which was defined as the ratio of static radial pressure to cyclic radial pressure explained less than 0.1% of the variation in wear or radial play.

7. We could not adapt the Rexnord equation to the twelve-bay test rig results.

8. The equation, wear = 60k PVT, can be used to approximate the wear life of the bearings tested on the twelve-bay test rig. By using the equations for k developed in this report, 32.8% and 63.2% of the variation in wear factor can be explained for the bearings tested without water and with water, respectively.

9. The results from two bearings tested on the four-bay test rig support the theoretical hypothesis that the wear life of a bearing is not affected by the sequence with which parts of a duty cycle are performed.

10. Interference fit was found to have no significant effect on the wear results obtained from the twelve-bay test rig.

RECOMMENDATIONS

The following recommendations are presented as a logical outcome of this contract and as a realization of the limitations on the wear-life equations as set forth in this report:

1. Additional testing should be done to improve the reliability of the wear-life equations.

2. Additional testing should be accomplished with bearings of other sizes and from other manufacturers to increase the scope of the design equations. As a minimum, the -16 size should be tested with a statistically valid number of specimens.

3. Testing should be done with out-of-plane motion in order to provide an improved equation which would allow considerations of this important and commonly occurring parameter. Note that this variable must be investigated in conjunction with the previously investigated variables. Extreme caution must be exercised in the experimental design planning stage in order to evaluate properly both the beneficial and the deleterious effects of out-of-plane motion. Out-of-plane motion can be beneficial by eliminating deleterious wear debris from the loaded interior of the bearing. Out-of-plane motion can cause accelerated wear by transferring debris-laden external surroundings into the interior of the bearing.

4. More testing should be done with water, a combination of water with sand and dust, and oil with sand and dust in order to obtain more information on these three important and commonly encountered wear initiators.

5. Consideration should be given to the use of an easily programmable, automatic loading and actuating system, such as an electrohydraulic actuated servo system with load and motion phasing capability, for future testing. (The increased accuracy in the loading and motions may help to reduce the standard error of the estimate and increase the correlation coefficient.)

6. The effect of different test rigs should be evaluated.

7. The bearing materials should be made less sensitive to water or water should be excluded from the ball-liner wear surface by a seal.

8. Additional continuous rotation tests should be performed with lower energy input levels in order to prove or disprove this low cost approach to evaluating bearing capacity. Also, additional tests should be performed to find out if total wear is independent of the sequence of a series of separate wear conditions.

9. The sample in Appendix E does not appear to be large enough to support the finding that size effects and manufacturing source (among the three best) have no significant effect on wear life. Also, Appendix E data shows no significant difference in wear life caused by water and wear life caused by sand and dust contamination, whereas this program identified significant difference in wear life. Therefore, more testing should be performed to expand the sample size in Appendix E.

APPENDIX A

DESCRIPTION OF FACTORIAL DESIGNED EXPERIMENT

RATIONALE FOR INITIAL SCREENING TESTS

Liner wear which has a direct influence on radial play can be affected by many factors in normal helicopter usage. Among these are:

1. Static radial pressure.

2. Cyclic radial pressure.

3. Speed of ball oscillation.

4. Ball oscillating angle.

5. Phase angle between the cyclic radial pressure and the ball oscillating angle.

6. Static axial load.

7. Cyclic axial load.

8. Fresh water contamination.

9. Dirt contamination.

10. Hydraulic fluid contamination.

11. Cleaning solvent contamination.

12. Salt water contamination.

13. Anti-icing fluid contamination.

14. Out-of-plane motion of the ball with respect to the liner (both transverse and rotational).

15. Ambient temperature.

16. Inner race (ball) or outer race (liner) rotation.

17. Cyclic radial pressure frequency with respect to ball oscillation frequency.

18. Phase angle between the cyclic axial load and the ball oscillating angle.

19. Cyclic axial load frequency with respect to ball oscillation frequency.

20. Bearing size.

21. Bearing manufacturer.

22. Ratio of static radial pressure to cyclic radial pressure (called load ratio).

23. Compression or tension loading.

24. Break-in.

25. Initial radial clearance.

26. Hours of usage.

27. Test rig differences.

28. Attitude (horizontal or vertical, shank up or down).

29. Bearing internal friction.

It is known that these factors do not act independently or entirely linearly in producing wear on a bearing, but rather act and interact in complex ways, the mechanisms of which are not thoroughly understood and for which analytical expressions do not exist.

Of these factors, factors 1 through 12, 14, 20 through 22, and 26 are considered to be the most significant. The investigation of factor 7 was ruled out because the expenditure of funds needed to modify the existing test rigs to develop cyclic axial loading would be excessive. Factor 12 was ruled out because Army helicopters normally do not operate in salt-laden air. Factor 13 was not tested because of the limited use of anti-icing fluid on helicopters.

Factor 14 can be deleterious to the wear life in cases where foreign contaminants are wiped into the wear zone and beneficial in cases where abrasive wear debris is wiped out of the wear zone. The investigation of this factor was ruled out because contract funds would not be sufficient to allow the necessary modifications to the existing test rigs. Factor 15 was not tested because high ambient temperatures (near 325°F) do not exist in normal helicopter usage and low temperatures (near -65°F), although classically causing high wear in Teflon bearings, have never caused degradation in actual cold-weather operation with helicopters. (The lack of low-temperature degradation in actual cold-weather operation may be the result of minimal extended operation at extremely low temperatures, bearing friction causing a temperature rise within the bearing in spite of the cold ambient conditions, or the low humidity associated with low temperature operation.)

Classically, inner race rotation causes higher wear on Teflon bearings than outer race rotation and has been attributed to the fact that inner race rotation subjects a smaller portion of the sacrificial liner material of the outer race to the wear process. Most helicopter bearings are subjected to this condition and for the sake of economy, it was decided that only inner race rotation would be used in this program.

Factors 20 and 21 are covered by the use of the Government-furnished data from tests on -6 and -16 bearing sizes from three bearing manufacturers. Factors 22, 23, and 26 are investigated in this program.

The remaining factors, 17 through 19, 24, 25, and 27 through 29, are considered to be of a lower order of importance or significance and have been ruled out in order to keep the cost of the test program within reasonable fiscal guidelines.

Given factors 1 through 6, 8 through 11, 22, 23, and 26, the design of a screening test was initiated. It was decided that factors 8 through 11 could be treated as one variable entitled contaminants and the variable would have five conditions: none; sand & dust; water; Skydrol 500A; and P-D-680, Type I, cleaning solvent.

Bearings used in controls which cycle only in response to a pilot input can be selected with relatively high confidence at this time. Control bearings, however, which exist in operating regimes where loads are generated by aerodynamic and inertial forces cannot be selected with a high degree of confidence. Therefore, the levels for factors 1 through 6 were chosen to encompass the typical loads and motions experienced by control bearings attached to rotating portions of flight controls.

Table A-1 lists the ranges of the variables experienced by various helicopters.

TABLE A-1

COMMON HELICOPTER RANGES OF VARIABLES

LOADS AND SPEEDS	AVERAGE CONTINUOUS	TRANSIENT
Static Radial Pressure	300 psi	1000 psi
Cyclic Radial Pressure	±200 psi	±2000 psi
Radial Angle of Oscillation	±5°	±10°
Transverse Angle of Oscillation	±3°	±6°
Rotational Angle of Oscillation	±3°	±6°
Cyclic Rates: 275 to 350 cpm, 1500 to 2000 cpm		
Surface Velocity	5 fpm	20 fpm
PV	20,000	40,000

Axial Static Load, pounds: 0% to 10% of Radial Static Load

Phase relationship of cyclic radial load with cyclic velocity: 0° to 90°

The following subparagraphs explain the choice of the levels for variables 1 through 6, 22, 23, and 26.

Static Radial Pressure (variable 1)

Usual values of static radial pressure are 300 psi with transient loadings to 1000 psi. MIL-B-81819, draft #5 specifies wear testing at a static radial pressure of 2000 psi with the rod end shank in compression. It was decided to use zero load, 1000 psi compression and 2000 psi compression. The existence of the 1000 psi compression load allows determination of the linearity of bearing wear with respect to static radial load. The use of 2000 psi rather than a lower maximum load allows comparisons with MIL-B-81819, draft #5 testing and also allows for future growth.

Compression or Tension Loading (variable 23)

There are indications from previous test work that rod end bearing wear can be affected by the direction of loading. Therefore, an additional level has been added to the static radial pressure variable: 2000 psi tension. The use of both a tension and compression load at the 2000 psi level allows determination of the difference in the two directions of loading and to what quantitative degree.

Speed of Ball Oscillation (variable 3)

A minimum level of 300 cpm was picked because this value is typical of the main rotor rpm for many operational helicopters. A maximum level of 900 cpm was chosen as being a reasonable upper limit for the speed capability of the test rig. The third level of 600 cpm allows determination of linearity of bearing wear with speed of ball oscillation.

Ball Oscillating Angle (variable 4)

Angles of 5, 10, and 15 degrees were chosen. The first two angles coincide with the angles usually seen in normal helicopter usage. The 15-degree angle provides the capability of developing a linear sliding velocity of 20 fpm (see Table A-1) even though the maximum speed of ball oscillation is slightly less than one-half the maximum encountered on the tail rotors of some types of helicopters.

Cyclic Radial Pressure (variable 2)

Levels of 0, 1000, 1500, and 2000 psi were chosen. This choice allows the linearity of bearing wear versus cyclic radial pressure to be determined. Also, these levels nicely cover the range of values listed in Table A-1.

Phase Angle Between the Cyclic Radial Pressure and the Ball Oscillating Angle (variable 5)

Levels of 0, 45, and 90 degrees were chosen and are defined in Figure A-1. This choice allows the linearity of bearing wear versus phase angle to be determined and covers the ranges of values listed in Table A-1.

Ratio of Static Radial Pressure to Cyclic Radial Pressure (Load Ratio) (variable 22)

The values of 0, 1000, and 2000 psi chosen for static radial pressure, when combined with the values of 0, 1000, 1500, and 2000 psi chosen for the cyclic radial pressure, produce ratios of 0, 1/2, 2/3, 1, 1-1/3, 2, and infinity. Previous test data indicated that roughly equal increments of wear can be caused by increasing the load ratio in successive increments from 0 to 0.3 to 1.0 to infinity. Therefore, the choices for the levels of static radial pressure and cyclic radial pressure provide an adequate coverage of the total range for the load ratio.

Static Axial Load (variable 6)

As shown in Table A-1, the axial static load can range from zero to 10% of the radial static load. The maximum static radial pressure chosen for testing is 2000 psi. Therefore, the maximum static radial load for the -6 test bearing equals 2000 psi multiplied by the ball diameter of 0.625 inch multiplied by the liner minimum width of 0.257 inch for a value of 321 pounds. Values of 0, 30, and 60 pounds were chosen for the static axial load, thus effectively bracketing the 10% of maximum radial static load value. The use of three levels also provides information on the linearity of the relationship between wear and static axial load.

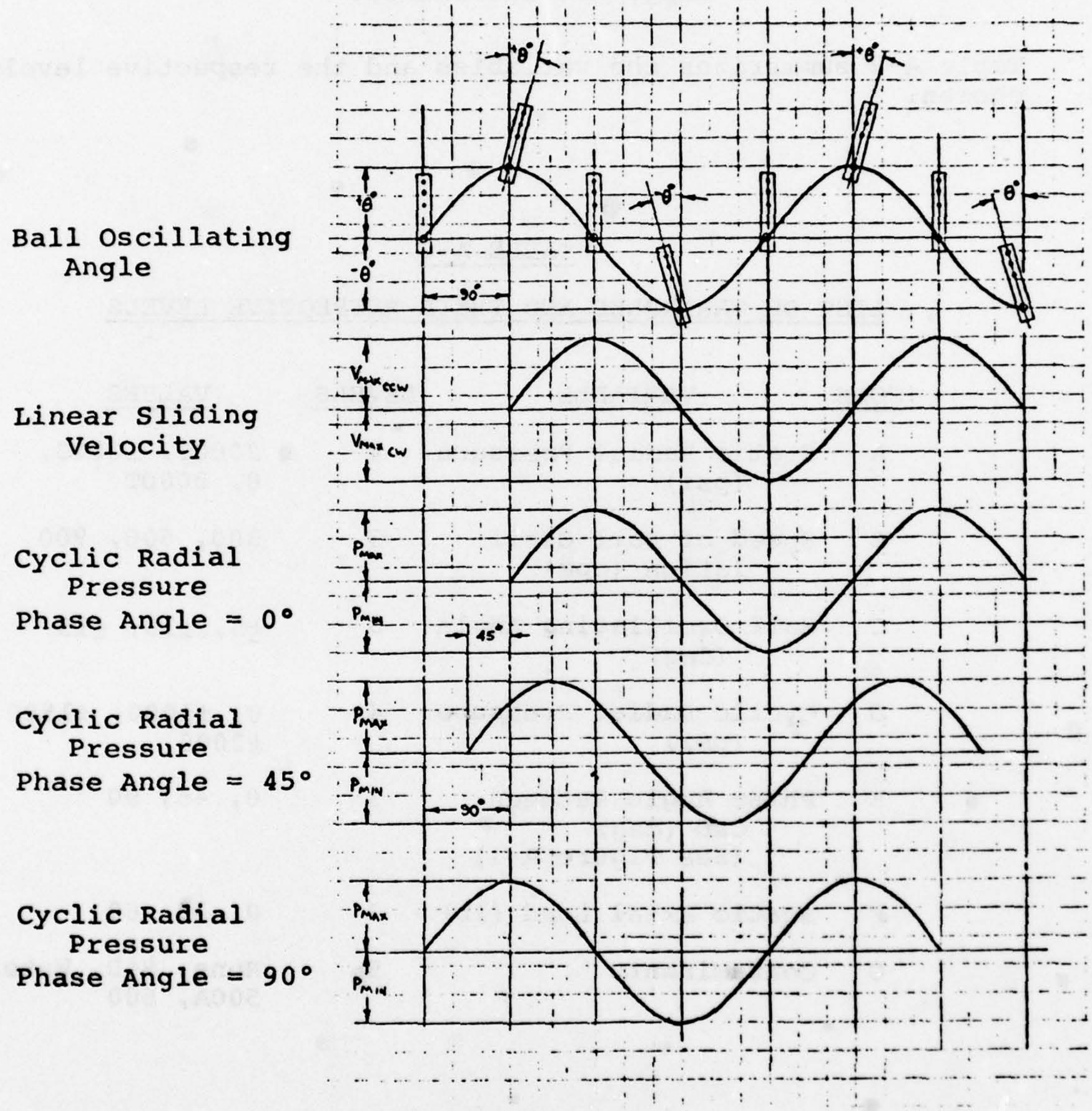


Figure A-1. Definition of Variable 5 - Phase Angle.

Hours of Usage (variable 26)

The screening tests consisted of 350 hours of test time for each test specimen unless failure occurred. From these tests and additional verification tests of 350-hour duration, the relationship between wear and hours of usage was determined.

Table A-2 summarizes the variables and the respective levels chosen.

TABLE A-2

LIST OF VARIABLES AND THEIR RESPECTIVE LEVELS

CODE	VARIABLE	LEVELS	VALUES
A	Static Radial Pressure (psi)	4	2000C, 1000C, 0, 2000T
B	Speed of Ball Oscillation (cpm)	3	300, 600, 900
C	Ball Oscillating Angle (deg)	3	±5, ±10, ±15
D	Cyclic Radial Pressure (psi)	4	0, ±1000, ±1500, ±2000
E	Phase Angle Between C&D (deg) (See Figure A-1)	3	0, 45, 90
F	Static Axial Load (lb)	3	0, 30, 60
G	Contaminants	5	None, S&D, Water, 500A, 680

SELECTION OF STATISTICAL TESTING TECHNIQUE

To examine, in a full factorial experiment, the main effects and all interactions for the seven variables and their respective levels as shown in Table A-2, a total of 4x3x3x4x3x3x5 (or 6480) runs would be required. The number of runs can be reduced to a practical size by some means of fractional replication. Various techniques are available, including fraction factorial, multiple balance, random assignment, response surface, and evolutionary operation.

In the range of ten variables, the usual choice is between multiple balance and random assignment. Multiple balance places limits on the number of levels for the several variables. These limitations might require more testing than is necessary. Response surface testing is a sequential design where the program can be redesigned after every test. It is useful when the use of a single piece of equipment is to be optimized. Evolutionary operation testing is a comparative process looking for a min/max answer. Each test series is followed by a repetition, again homing in on a min/max answer.

For the screening tests in the twelve-bay test rig, the technique used was random assignment, modified to eliminate meaningless combinations. For instance, phase angles of 45° or 90° are meaningless when the cyclic radial pressure is zero. Also, a static radial pressure of zero combined with a cyclic radial load of zero and a static axial load of zero is meaningless.

A summary of the major steps in the development of the experimental design is presented in the following pages.

INITIAL DESIGN OF EXPERIMENT

1. The variables were identified and were listed together with their number of levels and their values as shown in Table A-2. In so doing, the number of levels was determined by engineering judgement and the values were selected to represent the range of values encountered in typical helicopter operation.

2. The next order of business was to determine the number of test specimens to be used for the screening tests. One constraint on this decision was the fact that the existing twelve-bay wear test rig had to be used for the 350-hour screening tests as well as the 350-hour validation tests; thus, the total number of screening test specimens had to be some multiple of 12. The second constraint was a calendar time of approximately

5 months (108 working days) available to perform the screening tests and the validation tests. Based on a two-shift test operation, a 350-hour duration test requires 25 working days. Therefore, it was obvious that a maximum of three 350-hour screening tests could be performed and 36 test specimens would be needed. This number of specimens is more than adequate to provide the necessary results. Also, as will be seen later, the number 36 allows the use of 4 levels in lieu of the usual 3 levels for variables A and D.

3. Variables A, B, and C were chosen to be included in a three-factor full factorial design because the variables of static radial pressure, speed of ball oscillation, and ball oscillating angle make up the conventional "PV" variable. Thus, Design I, as shown in Figure A-2, provides a direct quantitative determination of the true significance of "PV". It is significant to note that the decision to use 36 test specimens for the screening tests was also based on the fact that variables A, B, and C were to be included in factorial design I with four levels for A and three levels each for B and C, thus requiring 36 test specimens to explore each one of the 36 possible combinations of A, B, and C.

4. A 36-specimen modified factorial design was established for variables D, E, and F as shown in Design II of Figure A-3. (The variables of cyclic radial pressure and phase angle were selected for Design II because phase angle and cyclic radial load are related by the definition of phase angle being the angle between the parameters of ball oscillation angle and cyclic radial pressure. The variable static axial load was selected for Design II partly by process of elimination and partly from engineering judgement that concludes that "pounding" due to cyclic radial load is minimized if axial loads are present.) Design II was not designed as a full factorial design and was purposely warped to eliminate testing at 45° or 90° phase angle when the cyclic radial load was zero. This was because phase angles of 45° and 90° are meaningless when cycle radial pressure is nonexistent.

5. The five contaminants were selected for the 36 test specimens in a randomized manner but at the same time heeding an arbitrary requirement that seven test specimens were to be tested with each of the contaminants and eight test specimens were to be tested with no contaminants.

6. The design control matrix shown in Figure A-4 was used to identify all possible first-order interactions. A total of 21 possible first-order interactions exist.

OSC. SPEED (cpm)	OSC. ANG. (deg)	STATIC RADIAL PRESSURE (psi) A1 2000C	A2 1000C	A3 0	A4 2000T
	C_1 5	11	12	7	3
B_1 600	C_2 10	2	5	6	4
	C_3 15	10	1	8	9
	C_1 5	16	18	23	19
B_2 900	C_2 10	14	21	13	22
	C_3 15	20	17	24	15
	C_1 5	35	31	33	30
B_3 300	C_2 10	25	34	28	32
	C_3 15	36	26	27	29

NOTE: Numbers 1 through 36 within the above array denote specimen identification.

Figure A-2. Design I.

PHASE ANG. (deg)		STATIC AXIAL LOAD (lb)		CYCLIC RADIAL PRESSURE (psi)			
				D_1 0	D_2 1000	D_3 1500	D_4 2000
E_1	0	F_1	0	12, 15	1		
		F_2	30	2, 9, 20, 21			28
		F_3	60	19, 34, 35		27	
E_2	45	F_1	0		25	3, 36	11, 30
		F_2	30		5, 23	6	29
		F_3	60		26	17	7
E_3	90	F_1	0		22	4, 33	8
		F_2	30		24	13	18
		F_3	60		16, 32	10	14, 31

Figure A-3. Design II.

	A	B	C	D	E	F	G
A							
B	X						
C	X	X					
D	X	X	X				
E	X	X	X	X			
F	X	X	X	X	X		
G	X	X	X	X	X	X	

Note: X - represents first-order interaction possibilities.

Figure A-4. Design control matrix.

7. An interdesign interaction matrix similar to the DxG interaction matrix of Figure A-5 was drawn for each of the 21 first-order interactions. Also, the interdesign interaction AxDxF was drawn as shown in Figure A-6.

8. Reassignment of the contaminants specified for the test specimens was made in order to improve the design of the experiment. (Note that the ideal design consists of an equal number of test specimens located in each element of each interdesign interaction matrix, but this condition is not always possible. As a minimum, there should be a number of specimens equal to one-half the quantity "N" where

$$N = \frac{\text{No. of Specimens}}{\text{No. of Elements in Interdesign Interaction Matrix}}$$

CYCLIC RAD. PRESS. (psi)	CONTAMINANTS				
	G_1 NONE	G_2 S&D	G_3 WATER	G_4 500A	G_5 680
D_1 0	2	9, 19	34, 35	12, 15	20, 21
D_2 1000	32	5, 24, 25	23, 26	16, 22	1
D_3 1500	4, 17, 36	13	3, 10	33	6, 27
D_4 2000	7, 29, 31	18	14	11, 28	8, 30

Figure A-5. Interdesign interaction DxG.

CYCLIC RAD. PRESS. (psi)	STATIC AXIAL LOAD (lb)	STATIC RADIAL PRESSURE (psi)			
		A_1 2000C	A_2 1000C	A_3 0	A_4 2000T
D_1 0	F_1 0		12		15
	F_2 30	2, 20	21		9
	F_3 60	35	34		19
D_2 1000	F_1 0	25	1		22
	F_2 30		5	23, 24	
	F_3 60	16	26		32
D_3 1500	F_1 0	36		33	3, 4
	F_2 30			6, 13	
	F_3 60	10	17	27	
D_4 2000	F_1 0	11		8	30
	F_2 30		18	28	29
	F_3 60	14	31	7	

Figure A-6. Interdesign interaction AxDxF.

Usually, it is good practice to provide at least 2 specimens per matrix element in order to provide replication. The AxDxF interdesign interaction matrix, however, has 48 elements and there are only 36 specimens available for test. Also, the constraint that zero cyclic radial load coupled with zero static radial load would result in inefficient usage of the available test specimens forces the AxD interdesign interaction matrix of Figure A-7 to have one unfilled element.)

CYCLIC RADIAL PRESS. (psi)	STATIC RADIAL PRESSURE (psi)			
	A_1 2000C	A_2 1000C	A_3 0	A_4 2000T
D_1 0	2, 20, 35	12, 21, 34		9, 15, 19
D_2 1000	16, 25	1, 5, 26	23, 24	22, 32
D_3 1500	10, 36	17	6, 13, 27, 33	3, 4
D_4 2000	11, 14	18, 31	7, 8, 28	29, 30

Figure A-7. Interdesign interaction AxD.

9. Finally, the choice of test bays and the order of testing (sequences) was randomized within the constraints of the system, resulting in the following table of test conditions (see Table A-3).

10. For reference purposes, Table A-4 lists the test specimens grouped into their respective load ratio categories and ranked in descending value of maximum PV.

TABLE A-3

TEST CONDITIONS FOR SCREENING TESTS IN TWELVE-BAY TEST RIG

SPECIMEN NO.	SEQUENCE NO.	BAY NO.	A STATIC RADIAL PRESS. (psi)	B SPEED OF OSC. (cpm)	C ANG. OSC. (deg)	D CYCLIC RADIAL PRESS. (psi)	E PHASE ANGLE (deg)	F STATIC AXIAL LOAD (lb)	G CONTAM.
1		5	1000C	600	15	1000	0	0	680
2		6	2000C	"	10	0	0	30	None
3	1	7	2000T	"	5	1500	45	0	Water
4		8	2000T	"	10	1500	90	0	None
5		5	1000C	"	10	1000	45	30	S&D
6		6	0	"	10	1500	45	30	680
7	2	7	0	"	5	2000	45	60	None
8		8	0	"	15	2000	90	0	680
9		5	2000T	"	15	0	0	30	S&D
10		6	2000C	"	15	1500	90	60	Water
11	3	7	2000C	"	5	2000	45	0	500A
12		8	1000C	"	5	0	0	0	500A
13		9	0	900	10	1500	90	30	S&D
14		10	2000C	"	10	2000	90	60	Water
15	3	11	2000T	"	15	0	0	0	500A
16		12	2000C	"	5	1000	90	60	500A
17		9	1000C	"	15	1500	45	60	None
18		10	1000C	"	5	2000	90	30	S&D
19	1	11	2000T	"	5	0	0	60	S&D
20		12	2000C	"	15	0	0	30	680
21		9	1000C	"	10	0	0	30	680
22		10	2000T	"	10	1000	90	0	500A
23	2	11	0	"	5	1000	45	30	Water
24		12	0	"	15	1000	90	30	S&D
25		1	2000C	300	10	1000	45	0	S&D
26		2	1000C	"	15	1000	45	60	Water
27	2	3	0	"	15	1500	0	60	680
28		4	0	"	10	2000	0	30	500A

TABLE A-3 (Continued)

TEST CONDITIONS FOR SCREENING TESTS IN TWELVE-BAY TEST RIG (Continued)

SPECIMEN NO.	SEQUENCE NO.	BAY NO.	A STATIC RADIAL PRESS. (psi)	B SPEED OF OSC. (cpm)	C ANG. OSC. (deg)	D CYCLIC RADIAL PRESS. (psi)	E PHASE ANGLE (deg)	F STATIC AXIAL LOAD (lb)	G CONTAM.
29		1	2000T	300	15	2000	45	30	None
30		2	2000T	"	5	2000	45	0	680
31	3	3	1000C	"	5	2000	90	60	None
32		4	2000T	"	10	1000	90	60	None
33		1	0	300	5	1500	90	0	500A
34		2	1000C	"	10	0	0	60	Water
35	1	3	2000C	"	5	0	0	60	Water
36		4	2000C	"	15	1500	45	0	None

TABLE A-4

SCREENING TEST CONDITIONS RANKED BY LOAD RATIO AND PV

SPECIMEN NO.	STATIC RADIAL LOAD (psi)a	CYCLIC RADIAL LOAD (psi)	LOAD RATIO (LR)b	SURFACE VELOCITY V (fpm)c	PV_{MAX} (psi-fpm)a,d	STATIC AXIAL LOAD (lb)	CONTAM.	PHASE ANG. (deg)
8	0	2000	0	16.36	±16,362	0	680	90
13	"	1500	"	16.36	±12,272	30	S&D	90
24	"	1000	"	24.54	±12,272	30	S&D	90
27	"	1500	"	8.18	±12,272	60	680	0
6	"	1500	"	10.91	±11,568	30	680	45
28	"	2000	"	5.45	±10,908	30	500A	0
7	"	2000	"	5.45	± 7,712	60	None	45
23	"	1000	"	8.18	± 5,784	30	Water	45
33	"	1500	"	2.73	± 2,045	0	500A	90
18	-1000	2000	1/2	8.18	-16,362	30	S&D	90
31	-1000	2000	1/2	2.73	- 5,454	60	None	90
17	-1000	1500	2/3	24.54	-50,572	60	None	45
14	-2000	2000	1	16.36	-49,087	60	Water	90
10	-2000	1500	1-1/3	16.36	-44,997	60	Water	90
22	2000	1000	2	16.36	40,906	0	500A	90
1	-1000	1000	1	16.36	-32,725	0	680	0
4	2000	1500	1-1/3	10.91	29,998	0	None	90
29	2000	2000	1	8.18	27,931	30	None	45
36	-2000	1500	1-1/3	8.18	-25,039	0	None	45
16	-2000	1000	2	8.18	-20,453	60	500A	90
5	-1000	1000	1	10.91	-18,620	30	S&D	45
11	-2000	2000	1	5.45	-18,620	0	500A	45
3	2000	1500	1-1/3	5.45	16,692	0	Water	45
25	-2000	1000	2	5.45	-14,764	0	S&D	45
26	-1000	1000	1	8.18	-13,965	60	Water	45
32	2000	1000	2	5.45	13,635	60	None	90
30	2000	2000	1	2.73	9,310	0	680	45

TABLE A-4 (Continued)

SCREENING TEST CONDITIONS RANKED BY LOAD RATIO AND PV (Continued)

SPECIMEN NO.	STATIC RADIAL LOAD (psi)a	CYCLIC RADIAL LOAD (psi)	LOAD RATIO (LR)b	SURFACE VELOCITY V (fpm)c	PV_{MAX} (psi-fpm)a,d	STATIC AXIAL LOAD (lb)	CONTAM.	PHASE ANG. (deg)
20	-2000	0	∞	24.54	-49,087	30	680	0
15	2000	"	"	24.54	-49,087	0	500A	"
9	2000	"	"	16.36	32,725	30	S&D	"
2	-2000	"	"	10.91	-21,817	30	None	"
19	2000	"	"	8.18	16,362	60	S&D	"
21	-1000	"	"	16.36	-16,362	30	680	"
34	-1000	"	"	5.45	- 5,454	60	Water	"
35	-2000	"	"	2.73	- 5,454	60	Water	"
12	-1000	"	"	5.45	- 5,454	0	500A	"

NOTES:
a. + Means Tension in Radial Load Arm
b. LR = Static Radial Pressure/Cyclic Radial Pressure
c. V = π (Ball Dia/12) (cpm) (4) (Ang.Osc./360)
d. PV_{MAX} = (Static Radial Pressure) (V) + (Cyclic Radial Pressure) (V) (c)

where c = 1 for 0° Phase Angle
c = .707 for 45° Phase Angle
c = .5 for 90° Phase Angle

APPENDIX B

STEPWISE MULTIPLE REGRESSION
(COMPUTER PROGRAM REGRESS)

INTRODUCTION

After the fractional factorial experiment had been designed and the test results had been obtained, various methods could have been used to obtain an equation for wear with respect to the defined independent variables. In one method, a person could perform analyses of variance to determine which variables have a statistically significant effect on the dependent variable, wear. A multiple regression analysis could then be performed to determine their mathematical relationships to the dependent variable. A particularly useful method lends itself well to this problem of equation development with multiple independent variables. This method is called stepwise multiple regression.

COMPUTER PROGRAM REGRESS

The stepwise multiple regression computer program, REGRESS, used for equation development on this contract was originally obtained from IBM and has been revised and improved by Kaman personnel. REGRESS is based on the abbreviated Doolittle method which is described in Appendix 6A of Reference 2.

The January 16, 1978 Kaman version of REGRESS has the capability of automatically performing a stepwise multiple regression analysis with any number of variables between 2 and 90 provided that the number of observations exceeds the number of variables by at least three. The program directs the IBM 360 computer to read the observations from IBM cards (12 observations per card), to read subset selection cards (one for each selection that is to be computed), and then to calculate the means, standard deviations, and the correlation coefficients for the total number of variables specified.

There is no program limitation on the number of selections that can be run. However, in each selection only one of the variables can be specified as the dependent variable and the

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2 Bennett, C.A., and Franklin, N.L., Statistical Analysis in Chemistry and the Chemical Industry, John Wiley and Sons, Inc., New York, N.Y., c. 1954.

remaining variables must be treated as independent variables. For each particular selection, any of the independent variables can be forced into or deleted from the regression equation, irrespective of its contribution to the equation. The program directs the computer to enter all the forced variables into the regression equation before all other independent variables. Within the set of forced variables, the computer is programmed to select the first variable to be entered as the one which explains the greatest amount of variance between it and the dependent variable. In other words, the variable with the highest partial correlation with the dependent variable is chosen first. The computer continues to step through all the forced variables, always picking the next variable to be entered as the one which explains the next greatest amount of variance, until there are no more forced variables. Then, the program directs the computer to step through the remaining independent variables, always picking the next variable to be entered as the one with the next highest partial correlation with the dependent variable.

At each step the computer calculates a regression equation based on the dependent variable and all the independent variables selected by all the steps up to and including the present step. The computer then prints out the following information before starting on the next step:

1. The sum of the squares reduced in this step.

2. The proportion of the sum of the squares reduced in this step.

3. The cumulative sum of the squares reduced in this step and all preceding steps.

4. The cumulative proportion of the sum of the squares reduced in this step and all preceding steps.

5a. The multiple correlation coefficient for the number of variables entered into the regression equation which is also the square root of the value in number 4 above.

5b. The same information as in 5a but adjusted for degrees of freedom.

6. The F-value for analysis of variance.

7a. The standard error of the estimate.

7b. The same information as in 7a but adjusted for degrees of freedom.

8. The regression coefficient, the standard error of the regression coefficient, and a computed t value for each of the variables entered.

9. The intercept of the regression equation, or in other words, the value of the dependent variable when all the entered variables are equated to zero.

Figure B-1 presents the computer printout for the third and fourth steps of a typical run.

Kaman's version of REGRESS also has the capability of data transformation and variable generation by computer. Thus, only the basic independent variables (such as static radial pressure, angle of oscillation, time, etc.) and the observation for the dependent variables (such as wear, radial play, etc.) have to be fed into the computer by use of IBM cards. One control card can be used to instruct the computer to generate needed variables or interactions between variables by performing any of the following arithmetic transformations:

1.	Addition	$X_{17} + X_{19}$
2.	Subtraction	$X_1 - X_2$
3.	Multiplication	$X_5 * X_{11}$
4.	Division	$X_6 \div X_9$
5.	Absolute value	$\lvert X_6 \rvert$
6.	Inverse	$1 \div X_{21}$
7.	Variable raised to a power	$(X_6)^3$
8.	Natural log of a variable	$\ln(X_{12})$
9.	Average of two variables	$(X_{14} + X_{15}) \div 2$
10.	Multiplication by a changing constant (dependent on value of another specified variable)	$X_9 * .707$ if $X_{11} = 45$ $X_9 * .5$ if $X_{11} = 90$ $X_9 * 1.0$ if $X_{11} = 0$

```
STEP   3

VARIABLE ENTERED.....33

SUM OF SQUARES REDUCED IN THIS STEP....   260890.375
PROPORTION REDUCED IN THIS STEP........        0.058

CUMULATIVE SUM OF SQUARES REDUCED......  2437564.000
CUMULATIVE PROPORTION REDUCED..........        0.541  OF  4502661.000

FOR  3 VARIABLES ENTERED
  MULTIPLE CORRELATION COEFFICIENT....    0.736
        (ADJUSTED FOR D.F.)...........    0.732
  F-VALUE FOR ANALYSIS OF VARIANCE...    69.248
  STANDARD ERROR OF ESTIMATE.........   108.321
        (ADJUSTED FOR D.F.)..........   108.931

  VARIABLE      REGRESSION       STD. ERROR OF      COMPUTED
   NUMBER       COEFFICIENT      REG. COEFF.        T-VALUE
     29            0.00169          0.00016          10.815
      7           38.05522          5.69677           6.680
     33           -0.04265          0.00905          -4.715
  INTERCEPT      -10.49914


STEP   4

VARIABLE ENTERED.....  8

SUM OF SQUARES REDUCED IN THIS STEP....   206394.375
PROPORTION REDUCED IN THIS STEP........        0.046

CUMULATIVE SUM OF SQUARES REDUCED......  2643958.000
CUMULATIVE PROPORTION REDUCED..........        0.587  OF  4502661.000

FOR  4 VARIABLES ENTERED
  MULTIPLE CORRELATION COEFFICIENT....    0.766
        (ADJUSTED FOR D.F.)...........    0.762
  F-VALUE FOR ANALYSIS OF VARIANCE...    62.233
  STANDARD ERROR OF ESTIMATE.........   103.055
        (ADJUSTED FOR D.F.)..........   103.934

  VARIABLE      REGRESSION       STD. ERROR OF      COMPUTED
   NUMBER       COEFFICIENT      REG. COEFF.        T-VALUE
     29            0.00161          0.00015          10.798
      7           38.38896          5.42055           7.082
     33           -0.04426          0.00861          -5.138
      8            0.87754          0.19907           4.408
  INTERCEPT      -78.92110
```

Figure B-1. Typical computer printout for program REGRESS.

11. Absolute value of sum $|X_{17} + X_{19}|$

12. Absolute value of quotient $|X_6 \div X_9|$

13. Change sign of variable A based on sign of variable B — If $X_6 = -$ Then $X_{18} = -$

14. Multiply variable by a factor of 10 — $(X_6 * 10^n)$

15. Divide variable by a factor of 10 — $(X_6 \div 10^n)$

16. Product of two variables divided by 1000 — $(X_6 * X_9) \div 1000$

17. Variable raised to a power n divided by $10^{(3n)}$ — $(X_6)^3 \div 10^9$

18. Absolute value of sum divided by 10^9 — $|X_{23} + X_{26}| \div 10^9$

19. Addition of a constant and a variable — $X_{88} + 46$

STATISTICAL VERIFICATION OF REGRESSION RESULTS

Throughout this program at least two checks were made to verify that the results were statistically significant. These checks are illustrated by using the computer print-out results of Figure B-1. The first method, shown in Table B-1, consists of comparing the multiple correlation coefficients obtained from program REGRESS with the minimum values for the multiple correlation coefficient (r) listed in Table B-2 for .05 and .01 levels of significance. Note that steps 1 and 2 were not presented in Figure B-1, but produced multiple correlation coefficients of 0.601 and 0.695, respectively. The critical value of r obtained in Table B-2 represents the numerical value which will be exceeded only 5 percent or 1 percent of the time for that sample size if the true correlation coefficient was zero. By comparison of the actual value of r with the critical value of r, we can either accept or reject the hypothesis that no association between the dependent variable and the independent variable exists ($r_{true} = 0$). If the actual value is less than the critical value, we conclude that no association exists. If greater, we conclude that a significant association exists.

TABLE B-1. TYPICAL SIGNIFICANCE CHECK FOR MULTIPLE CORRELATION COEFFICIENTS

STEP NUMBER	NO. OF VARIABLES	DEGREES OF FREEDOM	VALUES OF CORR. COEFFICIENT FROM TABLE B-2 $r_{.05}$	$r_{.01}$	CORR.COEFF.(r) OBTAINED IN REGRESS
1	2	178	.151	.197	.601
2	3	177	.188	.232	.695
3	4	176	.215	.256	.736
4	5	175	.236	.277	.766

The second method for significance testing consists of comparing the F-value for analysis of variance computed by REGRESS with the critical value obtained from a standard table of F-values. We illustrate this method for step 4 of Figure B-1. In order to obtain the critical F-value from a table of F-values, we must determine the degrees of freedom for the numerator, k, the degrees of freedom for the denominator, n-k-1, and the level of significance (see page 429 of reference 2). For step 4 of Figure B-1, k is 4 and n, the number of observations, is 180. Therefore, the critical F-value at the .005 level of significance is approximately 3.92. Thus, with the REGRESS calculated value of 62.2 being larger than the critical value, we can conclude that there is strong evidence, at the .005 level of significance, of a linear regression of the dependent variable upon the four independent variables.

TABLE B-2. TABLE OF .05 AND .01 PROBABILITY POINTS FOR r AND R

(Reproduced from "Tables for Statisticians" by Arkin and Colton, Barnes & Noble, Inc.)

Degrees of Freedom	NUMBER OF VARIABLES 2		3		4		5	
	.05	.01	.05	.01	.05	.01	.05	.01
1	.997	1.000	.999	1.000	.999	1.000	.999	1.000
2	.950	.990	.975	.995	.983	.997	.987	.998
3	.878	.959	.930	.976	.950	.983	.961	.987
4	.811	.917	.881	.949	.912	.962	.930	.970
5	.754	.874	.836	.917	.874	.937	.898	.949
6	.707	.834	.795	.886	.839	.911	.867	.927
7	.666	.798	.758	.855	.807	.885	.838	.904
8	.632	.765	.726	.827	.777	.860	.811	.882
9	.602	.735	.697	.800	.750	.836	.786	.861
10	.576	.708	.671	.776	.726	.814	.763	.840
11	.553	.684	.648	.753	.703	.793	.741	.821
12	.532	.661	.627	.732	.683	.773	.722	.802
13	.514	.641	.608	.712	.664	.755	.703	.785
14	.497	.623	.590	.694	.646	.737	.686	.768
15	.482	.606	.574	.677	.630	.721	.670	.752
16	.468	.590	.559	.662	.615	.706	.655	.738
17	.456	.575	.545	.647	.601	.691	.641	.724
18	.444	.561	.532	.633	.587	.678	.628	.710
19	.433	.549	.520	.620	.575	.665	.615	.698
20	.423	.537	.509	.608	.563	.652	.604	.685

TABLE B-2. TABLE OF .05 AND .01 PROBABILITY POINTS FOR r AND R (Cont)

Degrees of Freedom	NUMBER OF VARIABLES 2		3		4		5	
	.05	.01	.05	.01	.05	.01	.05	.01
22	.404	.515	.488	.585	.542	.630	.582	.663
24	.388	.496	.470	.565	.523	.609	.562	.642
26	.374	.478	.454	.546	.506	.590	.545	.624
28	.361	.463	.439	.530	.490	.573	.529	.606
30	.349	.449	.426	.514	.476	.558	.514	.591
35	.325	.418	.397	.481	.445	.523	.482	.556
40	.304	.393	.373	.454	.419	.494	.455	.526
45	.288	.372	.353	.430	.397	.470	.432	.501
50	.273	.354	.336	.410	.379	.449	.412	.479
60	.250	.325	.308	.377	.348	.414	.380	.442
70	.232	.302	.286	.351	.324	.386	.354	.413
80	.217	.283	.269	.330	.304	.362	.332	.389
90	.205	.267	.254	.312	.288	.343	.315	.368
100	.195	.254	.241	.297	.274	.327	.300	.351
200	.138	.181	.172	.212	.196	.234	.215	.253
500	.088	.115	.109	.135	.124	.150	.137	.162
1000	.062	.081	.077	.096	.088	.106	.097	.115

APPENDIX C

DETERMINATION OF VALIDATION TEST CONDITIONS (TWELVE-BAY TEST RIG)

Once the first three 350-hour tests in the twelve-bay test rig had been completed, the selection process for the test conditions for the validation tests could be started. First, a stepwise regression analysis was performed using the data from the 36 screening tests in order to obtain a preliminary equation for bearing wear. This equation is presented in Table C-1. The equation was then programmed into a time-share computer program entitled "CALC 1". This program instructed the computer to predict the wear after 350 hours of testing for all levels of the following primary variables: contamination, static radial pressure, CPM, angle of oscillation, cyclic radial pressure, static axial load, and phase angle. The total number of predicted wear calculations can be expressed as 5x5x3x3x5x3x3, or 10,125. The computer printout was then scanned and a total of 78 conditions which caused the highest and lowest wear were tabulated and are presented in Table C-2, column 10. A frequency distribution divided into 11 groups can be used to show the relative location of the 78 conditions and is presented in Figure C-1.

The selection of the twelve test conditions consisted of working with 3 matrices as shown in Figures C-2, C-3, and C-4. Initially, each matrix had 78 test condition designations located in their respective elements. Eventually, as the various conditions were eliminated, the matrices were reduced to their final appearance in Figures C-2, C-3, and C-4. It is important to note that a table of random numbers was used to eliminate bias in the selection of test conditions. The random numbers were also used to choose which of the four low speed test bays were to be used for each of the four low speed test conditions and similarly for the medium speed and high speed test bays. Table C-3 lists the test conditions that were used for the fourth 350-hour test. The following things can be noted about the conditions selected:

1. There are no duplications of conditions previously tested. (Case number 9a conditions are identical to those for specimen #35 which was tested in bay #3 in the first screening run, except that the contaminant is 500A, not water. By chance, even the same test bay will be used.)

2. With reference to Table C-2 and also the frequency distribution of Figure C-1, 7 test conditions produced low calculated wear (2 conditions were from

the A group, 4 from the B group, and 1 from the C group) and the remaining 5 test conditions produced high wear (4 from the L group and 1 from the K group).

3. Each of the five contamination levels is represented in the test conditions.

4. The various levels for the variables of static radial pressure, cpm, angle of oscillation, cyclic radial pressure, static axial load, and phase angle are well represented in the test conditions and are distributed quite well.

5. Tests with water contamination and zero static axial load now make up one-half of the tests with water contamination. Previously, only one out of the seven bearings tested with water had zero static axial load.

TABLE C-1

WEAR EQUATION OBTAINED FROM COMPUTER RUN LUBE 12F
(Designated as Equation LUBE 12F-2-20)

Wear = (Variable)(Reg.Coeff.)+Variable(Reg.Coeff.)+.......+ Intercept

VARIABLE NUMBER	VARIABLE DESCRIPTION	REGRESSION COEFFICIENT
16	Water (Time)	5.40272
11	Time	-4.93484
50	Water (Static Axial Load)	-22.04448
56	500A (Ang.Osc.) Static Axial Load	-0.11334
52	Absolute Value of Static Radial Pressure	-0.00537
10	Static Axial Load	20.85109
51	Water (Static Rad. Press.) Static Axial Load	-0.00503
31	Static Rad. Press (Static Axial Load)	0.00473
57	Water (Ang.Osc.) Static Axial Load	0.72971
35	Ang. Osc. (Static Axial Load)	-0.56232
5	Water	143.35999
40	Cyclic Rad. Press. (Static Axial Load) ÷1000	-2.90001
58	None (Cyclic Rad. Press) Static Axial Load ÷1000000	1078.06494
3	680	-24.74205
33	CPM (Static Axial Load)	-0.00221
41	CPM (Cyclic Rad. Press.) ÷1000	0.07697
38	Cyclic Rad. Press. (Phase Angle)	-0.00055
49	500A (Static Axial Load)	2.44135
2	Sand and Dust	20.41521
45	ln (Time)	-27.33029
	Intercept	4.27798

TABLE C-1 (Continued)

WEAR EQUATION OBTAINED FROM COMPUTER RUN LUBE 12F

DESCRIPTION OF VARIABLES

a. In variables 16, 50, 56, 51, 57, 5, 3, 49, and 2, substitute a "2" if the bearing has been contaminated with the specified contaminant (such as water) and a "1" if the bearing has not been contaminated.

b. In variable 58, substitute a "2" if the bearing has not been contaminated and a "1" if it has been contaminated.

c. Static radial pressure and cyclic radial pressure are in psi.

d. Angle of oscillation and phase angle are in degrees.

e. CPM is the cycles per minute of ball oscillation.

f. Static axial load is in pounds of force.

g. Time is in hours.

h. The total bearing wear calculated from this equation must be divided by 100,000 in order to obtain inches of wear.

TABLE C-2

CONDITIONS CAUSING THE HIGHEST AND LOWEST CALCULATED WEAR VALUES
(FROM EQUATION LUBE 12F-2-20)

CASE NO.	CONTAM.	S.R.P.	CPM	ANG. OSC.	C.R.P.	S.A.L.	PHASE ANG.	CALC. WEAR	NO. OF CONDITIONS	CLASS SYMBOL
1	5	3	3	1,3	5	1	1	2320	2	L
2	"	1,5	3	1,3	5	1	1	2310	4	L
3	"	2,4	2	1,3	5	1	1	2268	4	L
4	"	2,4	2	1,3	3	1	1	2222	4	K
5	"	3	2	2	4	1	2	2213	1	K
6	"	1,5	2	2	4	1	2	2203	2	K
7	"	1	2	2	4	3	2	542	1	D
8	5	2	2	1	5	3	3	538	1	D
9	4	5	1	1	1	3	1,3	336	2	C
10	4	3	1	1	1	3	1,3	311	2	"
11	2	3	3	1,3	5	1	1	306	2	"
12	4	5	1	3	1	3	1,3	300	2	"
13	2	1,5	3	1,3	5	1	1	295	4	"
14	1	3	3	1,3	5	1	1	285	2	"
15	4	4	2	1	1	3	1,3	284	2	"
16	1	1	3	1,3	5	1	1	275	2	"
17	3	3	3	1,3	5	1	1	261	2	"
18	2	2,4	2	1,3	5	1	1	254	4	C
19	3	1,5	3	1,3	5	1	1	250	4	B
20	4	2,4	2	1,3	1	3	1,3	248	4	"
21	4	5	2	2	2	3	2	235	1	"
22	1	2,4	2	1,3	5	1	1	234	4	"
23	3	2,4	2	1,3	5	1	1	209	4	"
24	5	1	1	1	5	3	3	207	1	"
*2	4	3	3	2	4	2	1	202	1	"
25	2	3	2	2	4	1	2	200	1	"
26	1	5	2	2	3	3	2	177	1	"
*1	2	1	1	2	1	2	1	172	1	"
27	3	3	2	2	4	1	2	154	1	B

TABLE C-2 (Continued)

CONDITIONS CAUSING THE HIGHEST AND LOWEST CALCULATED WEAR VALUES
(FROM EQUATION LUBE 12F-2-20)

CASE NO.	CONTAM.	S.R.P.	CPM	ANG. OSC.	C.R.P.	S.A.L.	PHASE ANG.	CALC. WEAR	NO. OF CONDITIONS	CLASS SYMBOL
28	1	1	2	2	4	3	2	93	1	B
29	4	1	2	2	4	3	2	75	1	"
30	1	2	2	1	5	3	3	39	1	"
31	2	1	2	2	4	3	2	17	1	"
32	1	1	1	1	5	3	3	9	1	B
33	4	2	2	3	5	3	3	-13	1	A
34	3	1	2	2	4	3	2	-29	1	"
35	4	1	1	3	5	3	3	-43	1	"
36	2	2	2	1	5	3	3	-70	1	"
37	2	1	1	1	5	3	3	-100	1	"
38	3	2	2	1	5	3	3	-115	1	"
39	3	1	1	1	5	3	3	-145	1	A

EXPLANATION OF CODE:

CONTAM.	STATIC RAD.PRESS.	CPM	ANG. OSC.	CYCLIC RAD. PRESS.	STATIC AX. LOAD	PHASE ANG.
1 None	1 2000T	1 300	1 5	1 0	1 0	1 0
2 S&D	2 1000T	2 600	2 10	2 500	2 30	2 45
3 680	3 0	3 900	3 15	3 1000	3 60	3 90
4 500A	4 1000C			4 1500		
5 Water	5 2000C			5 2000		

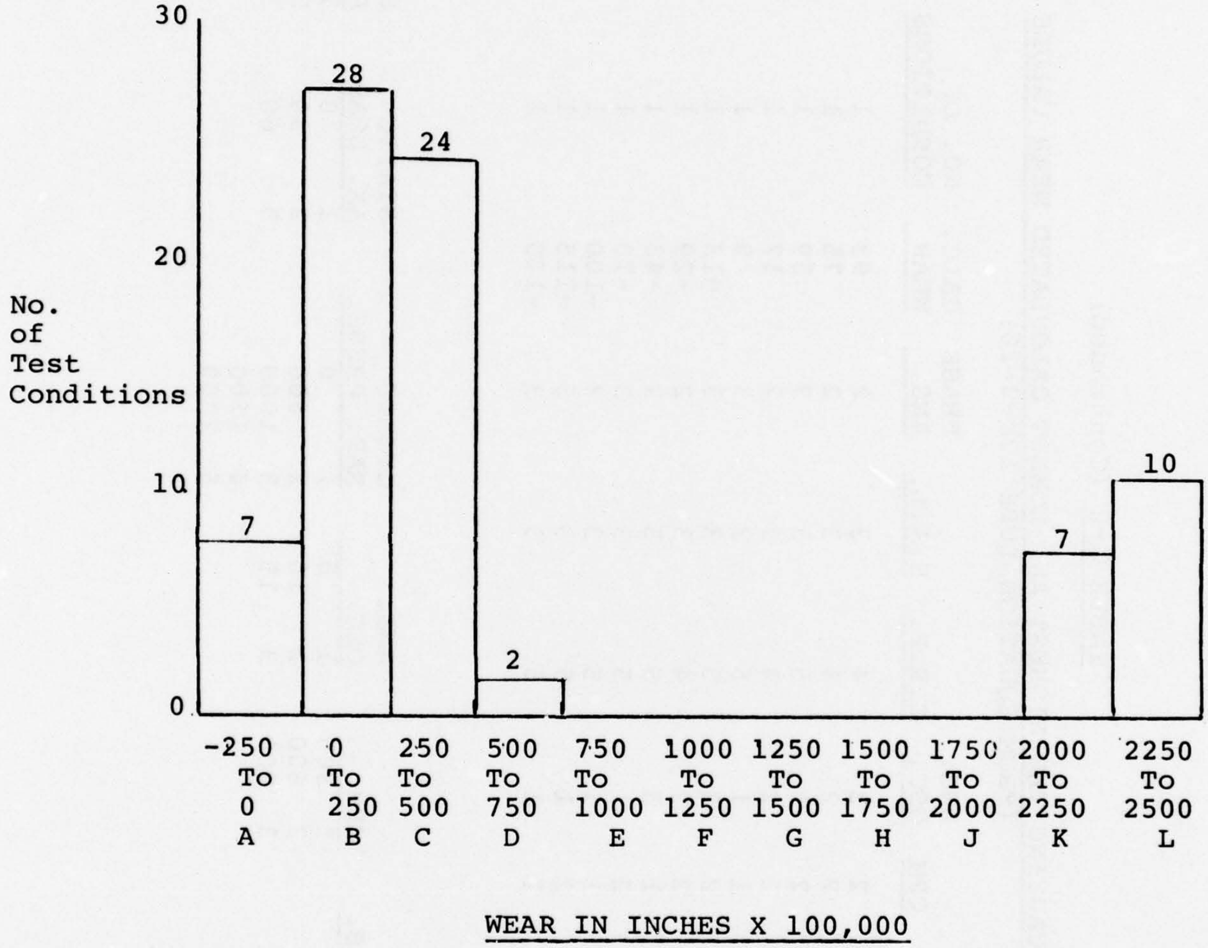

Figure C-1. Frequency distribution
(for test conditions producing highest and lowest wear).

Code		Code	S.R.P. → Ang. Osc.	1 2000T	2 1000T	3 0	4 1000C	5 2000C
1	300 CPM	1	5°	39				9a
		2	10°	*1				
		3	15°	35				
2	600 CPM		5°		3a		4c	
			10°			27		26
			15°					
3	900 CPM		5°	2a				
			10°			*2		
			15°			1b		2d

Figure C-2. Static radial pressure x speed of oscillation x angle of oscillation matrix.

Code		Code	C.R.P. → Static Axial Load	1 0	2 ±500	3 ±1000	4 ±1500	5 ±2000
1	0° Phase Ang.	1	0#			4c		1b,2a, 2d,3a
		2	30#	*1			*2	
		3	60#	9a				
2	45°		0#				27	
			30#					
			60#			26		
3	90°		0#					
			30#					
			60#					35,39

Figure C-3. Cyclic radial pressure x phase angle x static axial load matrix.

Code		No Water				
		1	2	3	4	5
		None	S&D	680	500A	Water
1	2000T		*1	39	35	2a
2	1000T					3a
3	0			27	*2	1b
4	1000C					4c
5	2000C	26			9a	2d

Figure C-4. Static radial pressure x contamination interaction matrix.

TABLE C-3

TEST CONDITIONS FOR FOURTH 350-HOUR TEST

CASE NO.	SPECIMEN NO.	BAY NO.	STATIC RAD. PRESS.	CPM	ANG. OSC.	CYCLIC RAD. PRESS.	PHASE ANGLE	STATIC AXIAL LOAD	CONTAM.
*1	37	1	2000T	300	10	0	0	30	S&D
35	38	2	2000T	"	15	2000	90	60	500A
9a	39	3	2000C	"	5	0	0	60	500A
39	40	4	2000T	300	5	2000	90	60	680
4c	41	5	1000C	600	5	1000	0	0	Water
3a	42	6	1000T	"	5	2000	0	0	Water
26	43	7	2000C	"	10	1000	45	60	None
27	44	8	0	600	10	1500	45	0	680
2a	45	9	2000T	900	5	2000	0	0	Water
*2	46	10	0	"	10	1500	0	30	500A
1b	47	11	0	"	15	2000	0	0	Water
2d	48	12	2000C	900	15	2000	0	0	Water

APPENDIX D

MODIFICATION OF REXNORD EQUATIONS FOR BEARING LIFE

INTRODUCTION

As discussed in the Deterministic Equation Development section of this report, attempts were made to modify existing wear equations to fit the test data obtained in this program. The results of the first attempt using a Rexnord equation are presented in this section of the report. The attempt with the best results utilized an equation based on PV and is presented in the Equation Development section of this report.

MODIFICATION OF REXNORD EQUATION

The Rexnord equation was presented in nomograph form in the Rexlon Engineering section of their Aerospace Bearings Catalog number 855, copyright 1969. The nomograph is presented in Figure D-1 and is restricted to those applications described by the following parameters:

1. Speed not exceeding 60 cycles per minute.
2. Effective pressures not exceeding 25,000 psi.
3. Oscillation up to 90° included angle (in other words, ±45°).

Even though the nomograph was obviously for low-speed, fixed-wing type bearing applications, it was felt that the equations could be modified to represent the high-speed, rotary-wing applications.

The following parameters were defined for the selection chart of Figure D-1 as follows:

1. $P_e = C_1 K_t (F_r + 5 F_a)$ where

   P_e = equivalent radial dynamic load, lb.
   C_1 = 1.2 (if bearing is directly exposed to water or deicers; otherwise, $C_1 = 1.0$).
   K_t = temperature adjustment factor.
   K_t = 1.0 (for temperatures up to 250°F).

   $K_t = 1.0 + \frac{(T-250)}{250}$ (for temperatures between 250°F and 500°F).
   T = maximum temperature anticipated in the application, °F.
   F_r = maximum applied dynamic radial load, lb.
   F_a = maximum applied dynamic axial load, lb.

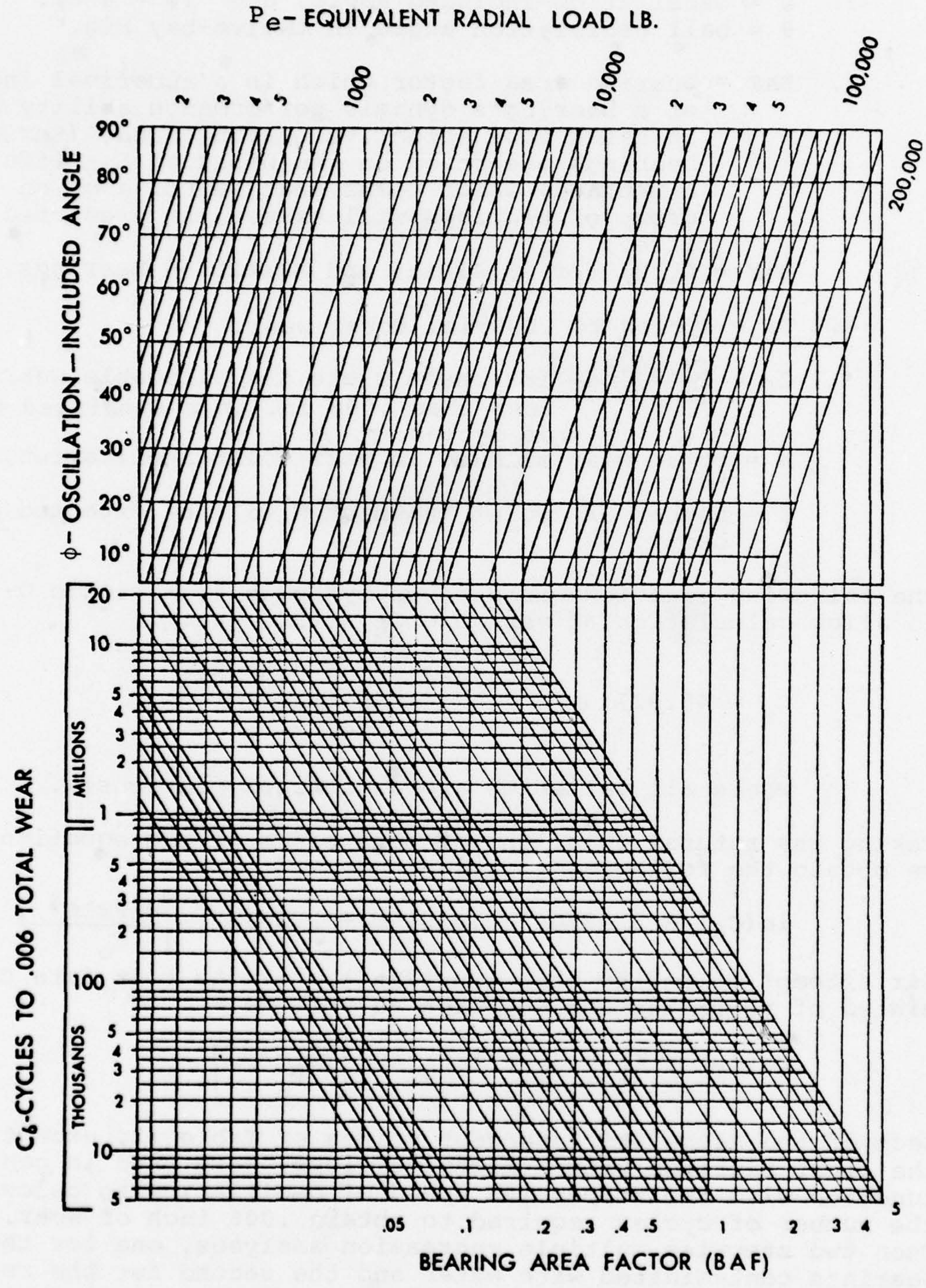

Figure D-1. Rexlon selection chart.

2. ϕ = oscillation-included angle, deg ($\phi = 2\ \theta$).
   θ = ball oscillation angle in twelve-bay rig.

3. BAF = bearing area factor which is a numerical index of a bearing's dynamic performance ability and is based upon effective bearing area. (Rexlon bearings are rated on the basis of L_{10} life expectancy which means that 90% of a given group of bearings will exceed the predicted life.)
   BAF = 0.181 for MS14101-6 and MS14104-6 bearings.

4. C_6 = cycles to .006 in. total wear.

   $C_6 = \frac{C(.0045)}{(W-.0015)}$ for cases where the allowable wear is less than .006 in. in a specified number of cycles.

   W = total wear allowed in particular application, in.

   C = number of cycles in which W will be expected to occur.

The following equation was derived by Kaman from Figure D-1 to allow calculation of variable C_6 within $\pm 4\%$:

$$C_6 = 45.9271\, e^{-.03619\phi} \left[\frac{BAF \times 10^6}{P_e}\right]^{2.3061}$$

where all variables are as defined previously.

Taking the natural logarithm of both sides of the equation, we obtain the following equation:

$$\ln(C_6) = 3.82706 - 0.03619\phi + 2.3061 \ln\left[\frac{BAF \times 10^6}{P_e}\right]$$

Our attempt to modify this equation to fit the test data consisted of rewriting the equation as follows:

$$\ln(C_6) = K_1 + K_2\ \phi + K_3 \ln\left[\frac{BAF \times 10^6}{P_e}\right]$$

Each of the slope and intercept values of Table 12, except for the eight bearings with a negative slope, were used in conjunction with the respective speed of oscillation to calculate the number of cycles required to obtain .006 inch of wear. Then two stepwise multiple regression analyses, one for the 12 bearings contaminated with water and the second for the remaining 28 bearings without water contamination, were performed using $\ln(C_6)$ as the dependent variable. The independent variables were ϕ and $\ln\left[\frac{BAF\times10^6}{P_e}\right]$. Coefficients K_1, K_2, and K_3 were to be calculated by the regression program. However, in both analyses the regression program automatically aborted the computation after only two of the coefficients had been determined. The resultant error message stated that computation of the third coefficient required the impossible task of obtaining the square root of a negative number. Multiple correlation coefficients of .157 and .132 were obtained for the water contaminated and non-water-contaminated bearings, respectively.

Two subsequent stepwise multiple regression analyses resulted in successful computation of all three coefficients. Multiple correlation coefficients of 0.431 and 0.377 were obtained for the water contaminated and non-water-contaminated bearings, respectively. This was accomplished by revising Rexnord's equivalent radial dynamic pressure parameter to incorporate our phase angle variable as shown below:

$$P_{em} = C_1\, K_t\, [F_{r\phi} + 5\, F_a] \text{ where}$$

P_{em} = modified equivalent radial dynamic load, lb

$F_{r\phi} = |F_s|+|F_c|$ if $X_{13} = 0°$

$F_{r\phi} = |F_s|+|0.707\, F_c|$ if $X_{13} = 45°$

$F_{r\phi} = |F_s|+|0.5\, F_c|$ if $X_{13} = 90°$

F_s = static radial load, lb

F_c = cyclic radial load, lb

X_{13} = phase angle between cyclic radial load and ball oscillation angle, deg

$F_r = |F_s|+|F_c|$ (in the unmodified Rexnord equation).

The results of the two regression analyses are presented below:

	12 bearings with water contamination	28 bearings without water contamination
r	= 0.431	= 0.377
r^2	= 0.186	= 0.142
K_1	= 10.61767	= 14.34051
K_2	= -0.00324	= -0.01668
K_3	= 0.81798	= 0.76048
S_y	= 0.674	= 1.046
C_v	= 4.41%	= 5.61%
S_y/σ	= 0.997	= 0.963

Thus, the modified Rexnord equations are:

<u>Bearings with water contamination</u>

$$C_6 = (0.04085 \times 10^6)e^{-.00324\phi}\left[\frac{BAF\times10^6}{P_{em}}\right]^{0.81798} \pm 3.32\times10^6$$

<u>Bearings without water contamination</u>

$$C_6 = (1.6905 \times 10^6)e^{-.01668\phi}\left[\frac{BAF\times10^6}{P_{em}}\right]^{0.76048} \pm 13.3\times10^7$$

These equations explain only 18.6% and 14.2% of the total variability in the variable C_6 with water and without water contamination, respectively. Thus, it was concluded that we cannot adapt the Rexnord equation to the test conditions and results reported herein.

APPENDIX E

ANALYSIS OF FURNISHED DATA

The Naval Air Development Center at Warminster, Pa. has performed tests on bearings from eight bearing suppliers. Wear data from the best three out of eight suppliers were furnished by the Government for use in this contract. The steady-load dry-lubricated bearing wear data were obtained from tests performed in accordance with MIL-B-81819, Draft #5. Both -6 and -16 bearings are represented in the data. Data from twelve -6 bearings were received and consisted of four from each of three manufacturers with two of the four tested in water and the other two tested in sand and dust. The remaining 18 bearings were -16 bearings consisting of six bearings from each of the same three manufacturers with three of the six tested in water and the other three in sand and dust. The bearings were identified by code letters A, B, and C but no information was given as to which manufacturer was manufacturer A, etc.

An analysis of variance was performed on the bearing data in order to determine if there were significant effects on the wear caused by size, manufacturer, or contaminant. Two methods were used to analyze the data in an analysis of variance program. The first method used the slopes of the wear curves as the input data. The second method used the number of hours required to reach 0.005 inch of wear. In both methods the analysis of variance program indicated that there is no significant effect on wear caused by size, manufacturer, or contaminant to the .05 level of significance. Based upon our test results from the twelve-bay test rig, water has a significant effect on wear life. We recommend that additional bearings be tested to increase the population of tested bearings and consequently improve the confidence level of the analysis of variance.

APPENDIX F

INTERFERENCE FIT, INITIAL RADIAL PLAY, AND INITIAL AXIAL PLAY

Table F-1 presents a tabulation of the interference fit, initial radial play, and initial axial play for all 48 bearings tested in the twelve-bay test rig. Although there was no requirement to investigate these factors, it was decided to enter them in the stepwise regression analyses. Interference fit, initial radial play, and initial axial play were entered as variables 16, 17, and 18, respectively. None of these factors were found to have any significant effect on test bearing wear. This could have been expected for the initial radial play and initial axial play parameters because these bearings were so tight that the radial play and axial play were unusually low. However, the randomness of the specimen selection process did provide a good distribution of interference fit as shown in Figures F-1 and F-2. Engineering judgement leads one to believe that initial radial play and initial axial play would have had a significant effect, if not on bearing wear, then on radial play. This program leaves this question unanswered since there were no relatively loose bearings available for testing.

TABLE F-1

TEST BEARING INTERFERENCE FITS AND INITIAL PLAY READINGS

RUN NO.	BAY NO.	BEARING S/N	INTERFERENCE FIT(IN.x100,000)	INITIAL PLAY (IN.x100,000) RADIAL PLAY	AXIAL PLAY
1	1	DL-13	25	30	5
"	2	DL-14	35	20	20
"	3	DL-11	50	30	5
"	4	DL-12	35	25	25
"	5	DL-9	60	0	10
"	6	DL-10	25	25	10
"	7	DL-7	40	0	30
"	8	DL-8	30	20	10
"	9	DL-1	58	0	10
"	10	DL-3	60	0	10
"	11	DL-5	20	15	0
"	12	DL-6	45	15	25
2	1	DL-28	25	25	10
"	2	DL-26	10	35	0
"	3	DL-18	55	60	0
"	4	DL-19	35	45	25
"	5	DL-25	60	30	20
"	6	DL-20	50	20	20
"	7	DL-24	40	15	0
"	8	DL-22	40	20	5
"	9	DL-21	38	10	10
"	10	DL-23	60	75	0
"	11	DL-27	45	20	0
"	12	DL-29	40	35	20
3	1	DL-37	55	10	20
"	2	DL-32	40	0	2.5
"	3	DL-38	80	5	10
"	4	DL-39	60	0	15
"	5	DL-40	95	15	5
"	6	DL-41	80	60	20
"	7	DL-33	75	15	20
"	8	DL-34	30	30	10
"	9	DL-42	62	10	0
"	10	DL-43	90	45	25
"	11	DL-35	40	15	0
"	12	DL-36	80	25	5
4	1	DL-52	30	15	30
"	2	DL-44	40	30	15
"	3	DL-45	60	15	15
"	4	DL-46	45	0	5
"	5	DL-49	85	20	10
"	6	DL-50	60	55	20
"	7	DL-55	55	15	20
"	8	DL-47	50	25	0
"	9	DL-54	57	45	0
"	10	DL-48	90	20	20
"	11	DL-53	60	0	0
"	12	DL-51	60	10	35

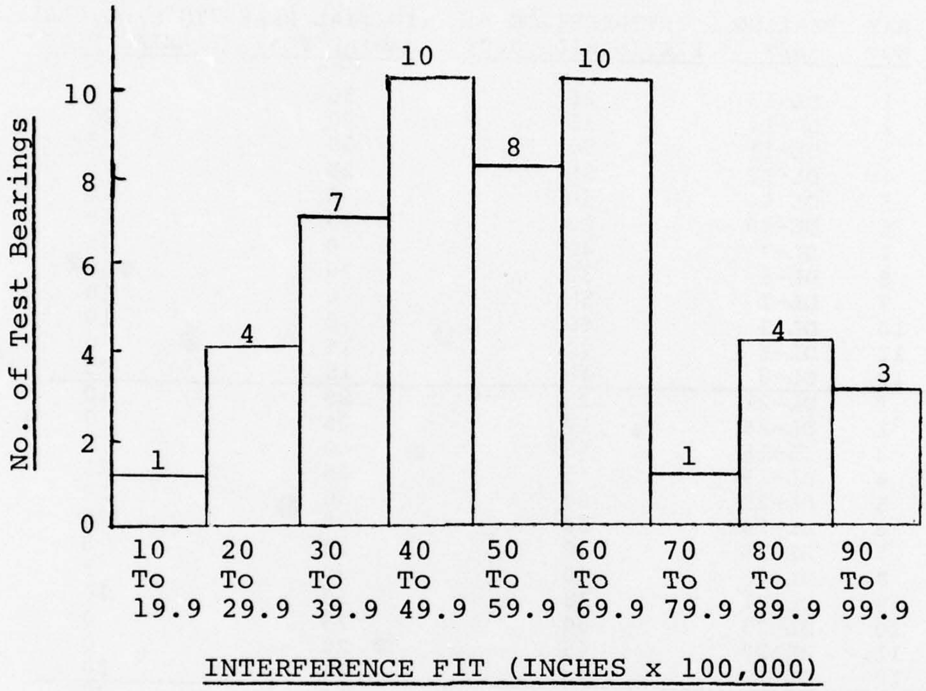

Figure F-1. Frequency distribution of test bearing interference fit.

INTER-FERENCE FIT ($INx10^5$)	STATIC RADIAL PRESSURE (psi)				
	2000T	1000T	0	1000C	2000C
10 To 19.9				1*	
20 To 29.9	1		1		2
30 To 39.9	2		1	3	1
40 To 49.9	5		4		1
50 To 59.9	2		3	1	2
60 To 69.9	2	1	2	3	2
70 To 79.9					1
80 To 89.9				2	2
90 To 99.9	1		1		1
TOTAL	13	1	12	10	12

*Numbers in Matrix denote number of test bearings.

Figure F-2. Interference fit - static radial pressure distribution.

APPENDIX G

VARIABLES USED IN STEPWISE MULTIPLE REGRESSION PROGRAM FOR EMPIRICAL DETERMINATION OF WEAR-LIFE EQUATIONS

Table G-1 lists the 76 variables which were used to develop the four wear equations presented in Figure 29. Note that variables 14, 15, and 49 were deleted from the regression when variable 12 was specified to be the dependent variable. Similarly, variables 12, 15, and 49 were deleted when variable 14 was the dependent variable. The remaining 72 variables were eligible for selection by the computer as described in Appendix B. Those that do not appear in the equations of Figure 29 were deleted by the computer because they did not explain more than 0.2% of the variation in the dependent variable.

TABLE G-1. LIST OF VARIABLES USED FOR EQUATIONS SHOWN IN FIGURE 29

VAR #	! VAR NAME
1	NONE
2	S&D
3	P-D-680
4	SKYDROL 500A
5	WATER
6	STATIC RAD PRES
7	SPEED
8	ANG OSC
9	CYCLIC RAD PRES
10	STATIC AX LOAD
11	TIME
12	TOTAL BRG WEAR
13	PHASE ANG
14	COMP WEAR
15	TENS WEAR
16	INTERF FIT
17	INITIAL RD PLAY
18	INITIAL AX PLAY
21	WATER*TIME
22	LOAD RATIO
23	CPM*ANG OSC
25	CYCLIC LOAD
26	+OR-CYCLIC LD
27	CYC WORK/1000
28	ABS(PV)/10^12
29	((SRP)^2)/10^6
30	SRP*ANG OSC
32	MOD LD RATIO
33	(X6*X7)/1000
34	X5*X6*X8
35	X6*X13
36	X6*X10
37	X5*X6
38	X7*X10
39	X7*X13
40	X8*X10
41	X5*X13
42	(X10*X13)/1000
43	(X9*X13)/1000
44	(X6*X9)/1000
45	(X9*X10)/1000
46	(X7*X9)/1000
47	X8*X9
48	(X8*X13)/1000
49	RADIAL PLAY
50	LOGN(TIME)
51	((SRP)^3)/10^9
52	X2*X6*X10/1000
53	X3*X10*X13/1000
54	X4*X10
55	X5*X10
56	X5*X6*X10/1000
57	ABS VAL(SRP)
58	MAX STRESS
59	MIN STRESS
60	STRESS RANGE
61	X3*X10
62	X5*X8*X10/1000
63	X2*X10
64	X5*X9
65	X4*X9
66	X3*X9
67	X2*X9
68	X4*X6
69	X3*X8
70	X3*X13
71	(X3*X26)/1000
72	X3*X6
73	X4*X8
74	X4*STA WRK/10^6
75	X2*X6
76	X5*X7*X8
77	X4*X7*X8
78	X3*X7*X8
81	(STAT WRK)/10^6
88	(X9/X10)/1000

TABLE G-1 (Continued). LIST OF VARIABLES USED FOR EQUATIONS SHOWN IN FIGURE 29.

Notes:

1. Variables 1 through 5 are the contaminants listed in Table 4.
2. Variables 6 through 10 and 13 are the variables listed in Table 4.
3. Variables 11 and 50 are time in hours and the natural logarithm of time, respectively.
4. Variables 12, 14, and 15 are the wear values listed in Tables 9, 7, and 8, respectively.
5. Variables 16, 17, and 18 are interference fit, initial radial play, and initial axial play as described in Appendix F.
6. Variables with numbers higher than 18 were calculated by the computer as explained in Appendix B.
7. The symbols * and / indicate multiplication and division, respectively.
   Example: Variable 56 equals water multiplied by static radial pressure multiplied by static axial load divided by 1000.
8. Variable 22 is load ratio which was calculated by obtaining the absolute value of static radial pressure divided by cyclic radial pressure.
9. Variable 25 is cyclic load which was calculated by multiplying cyclic radial pressure by 1.0, 0.707, or 0.5 when the phase angle equals 0°, 45°, or 90°, respectively.
10. Variable 26 is the same magnitude as variable 25 except variable 26 assumes the same sign as variable 6. (If variable 6 is tension, then variable 26 will be tension.)
11. Variable 27 is cyclic work divided by 1000 and is variable 23 multiplied by variable 26 and divided by 1000.
12. Variable 81 is static work divided by 10^6 and is variable 23 multiplied by variable 6 and divided by 10^6.
13. Variable 28 is the absolute value of PV divided by 10^{12}. It is the absolute value of the sum of variable 81 multiplied by 1000 and variable 27.
14. Variable 29 is the square of static radial pressure divided by 10^6.
15. Variable 32 is modified load ratio and equals variable 6 divided by variable 26.
16. Variable 49 is radial play and equals the summation of variables 14 and 15.
17. Variable 51 is the cube of static radial pressure divided by 10^9.

TABLE G-1 (Continued). LIST OF VARIABLES USED FOR EQUATIONS SHOWN IN FIGURE 29.

18. Variable 57 is the absolute value of static radial pressure.
19. Variable 58 is the summation of variable 6 and variable 9.
20. Variable 59 is variable 6 minus variable 9.
21. Variable 60 is cyclic radial pressure multiplied by 2.
22. Variable 74 is variable 4 multiplied by variable 81.
23. To avoid possible divisions by zero, a 1 was used for all zero values of variables 6 and 9. Also, 0.1 was used for all zero values of variables 10 and 13.

APPENDIX H

DESCRIPTION OF EQUATION DEVELOPMENT FOR LINER WEAR ON COMPRESSION SIDE, LINER WEAR ON TENSION SIDE, AND RADIAL PLAY

Equation LUBE 24A-1-6 for liner wear on the compression side, equation LUBE 24G-1-7 for liner wear on the tension side, and LUBE 25A-1-9 for radial play were developed from the twelve-bay test rig data. Table H-1 lists thirty-five test bearings and twenty-nine test bearings which were used for the liner wear equations for wear on the compression side and wear on the tension side, respectively. It can be noted that some of the bearings are used for both equations because these particular bearings were subjected to both tension and compression during one complete cycle of ball oscillation. As shown in Table H-1, three bearings from the compression load group and three from the tension load group were not used for equation development but instead for equation confirmation.

Table H-2 lists the thirty-five test bearings used for the radial play equation development and the four test bearings used for equation confirmation. Table H-2 also lists the nine test bearings which could not be used for equation development because the negative wear readings had not been measured.

Tables H-3, H-4, and H-5 present the test balance for the three equations. Table H-6 presents the comparison of actual and predicted wear values for the three confirming tests for each of the two liner wear equations. From the Residual/Sy values in Table H-6 we find that the equation for wear on the compression side predicts only one out of three confirming tests within ±3 Sy. The equation for wear on the tension side predicts two out of three confirming tests within ±3 Sy.

Table H-7 presents the comparison of actual and predicted radial play values for the four confirming tests for the radial play equation. The radial play equation predicts three out of four confirming tests within ±3 Sy.

Values of ±3 standard errors of the estimate are conventionally used confidence limits for verifying the ability of regression equations to predict actual outcomes. In a Gaussian distribution, 99.7% of all the values will fall within these limits. When actual results come outside of these limits, it indicates a most unusual occurrence or that the equation does not predict well. Therefore, it is concluded the predictive equations do not predict well for the very short lives that result when water is present.

TABLE H-1. LIST OF TEST BEARINGS USED FOR EQUATION DEVELOPMENT AND EQUATION CONFIRMATION

EQUATION DEVELOPMENT

W_C EQUATION, LUBE 24A-1-6		W_T EQUATION, LUBE 24G-1-7	
BEARING DESIGNATION		BEARING DESIGNATION	
DL-13 (1-1)	DL-22 (2-8)	DL-13 (1-1)	DL-38 (3-3)
DL-14 (1-2)	DL-21 (2-9)	DL-7 (1-7a)	DL-39 (3-4)
DL-11 (1-3)	DL-27 (2-11)	DL-8 (1-8)	DL-40 (3-5)
DL-12 (1-4)	DL-29 (2-12)	DL-1 (1-9)	DL-42 (3-9)
DL-9 (1-5)	DL-38 (3-3)	DL-3 (1-10)	DL-35 (3-11)
DL-10 (1-6)	DL-41 (3-6)	DL-5 (1-11)	DL-52 (4-1)
DL-1 (1-9)	DL-33 (3-7)	DL-18 (2-3)	DL-44 (4-2)
DL-3 (1-10)	DL-34 (3-8)	DL-19 (2-4)	DL-50 (4-6)
DL-6 (1-12)	DL-42 (3-9)	DL-20 (2-6)	DL-47 (4-8)
DL-28 (2-1)	DL-43 (3-10)	DL-24 (2-7)	Dl-48 (4-10)
DL-26 (2-2)	DL-36 (3-12)	DL-22 (2-8)	
DL-18 (2-3)	DL-50 (4-6)	DL-23 (2-10)	
DL-19 (2-4)	DL-55 (4-7)	DL-27 (2-11)	
DL-25 (2-5)	DL-47 (4-8)	DL-29 (2-12)	
DL-20 (2-6)	DL-48 (4-10)	DL-37 (3-1)	
DL-24 (2-7)	DL-51 (4-12)	DL-32 (3-2)	

EQUATION CONFIRMATION

W_C EQUATION, LUBE 24A-1-6	W_T EQUATION, LUBE 24G-1-7
BEARING DESIGNATION	BEARING DESIGNATION
DL-45 (4-3)	DL-46 (4-4)
DL-49 (4-5)	DL-54 (4-9)
DL-53 (4-11)	DL-53 (4-11)

TABLE H-2. LIST OF TEST BEARINGS USED FOR EQUATION DEVELOPMENT AND EQUATION CONFIRMATION

EQUATION DEVELOPMENT

R.P. EQUATION, LUBE 25A-1-9

BEARING DESIGNATION			
DL-13 (1-1)	DL-18 (2-3)	DL-37 (3-1)	DL-52 (4-1)
DL-12 (1-4)	DL-19 (2-4)	DL-32 (3-2)	DL-44 (4-2)
DL-9 (1-5)	DL-25 (2-5)	DL-38 (3-3)	DL-49 (4-5)
DL-7 (1-7a)	DL-20 (2-6)	DL-39 (3-4)	DL-50 (4-6)
DL-8 (1-8)	DL-24 (2-7)	DL-41 (3-6)	DL-55 (4-7)
DL-1 (1-9)	DL-22 (2-8)	DL-33 (3-7)	DL-47 (4-8)
DL-3 (1-10)	DL-23 (2-10)	DL-42 (3-9)	DL-48 (4-10)
DL-28 (2-1)	DL-27 (2-11)	DL-43 (3-10)	DL-51 (4-12)
DL-26 (2-2)	DL-29 (2-12)	DL-36 (3-12)	

EQUATION CONFIRMATION

R.P. EQUATION, LUBE 25A-1-9

BEARING DESIGNATION			
DL-45 (4-3)	DL-46 (4-4)	DL-54 (4-9)	DL-53 (4-11)

BEARINGS NOT USED BECAUSE NEGATIVE WEAR READINGS WERE NOT OBTAINED (SEE TABLES 7 AND 8)

BEARING DESIGNATION			
DL-14 (1-2)	DL-5 (1-11)	DL-21 (2-9)	DL-34 (3-8)
DL-11 (1-3)	DL-6 (1-12)	DL-40 (3-5)	DL-35 (3-11)
DL-10 (1-6)			

TABLE H-3. SUMMARY OF TEST BALANCE
(FOR W_C EQUATION, LUBE 24A-1-6)

CODE	VARIABLE	LEVEL	NUMBER OF SPECIMENS 32 TESTS	3 TESTS
A	Static Radial Pressure (psi)	2000C	11	1
		1000C	9	1
		0	11	1
		1000T	1	0
		2000T	0	0
B	Speed of Ball Oscillation (cpm)	300	9	1
		600	12	1
		900	11	1
C	Ball Oscillating Angle (deg)	5	10	2
		10	12	0
		15	10	1
D	Cyclic Radial Pressure (psi)	0	6	1
		1000	8	1
		1500	9	0
		2000	9	1
E	Phase Angle Between C&D (deg)	0	12	3
		45	11	0
		90	9	0
F	Static Axial Load (lb)	0	10	2
		30	11	0
		60	11	1
G	Contaminants	None	6	0
		S&D	5	0
		680	7	0
		500A	6	1
		Water	8	2
-	Predicted Wear	Low	--	1
		High	--	2

TABLE H-4. SUMMARY OF TEST BALANCE
(FOR W_T EQUATION, LUBE 24G-1-7)

CODE	VARIABLE	LEVEL	NUMBER OF SPECIMENS 26 TESTS	3 TESTS
A	Static Radial Pressure (psi)	2000C	0	0
		1000C	3	0
		0	11	1
		1000T	1	0
		2000T	11	2
B	Speed of Ball Oscillation (cpm)	300	9	1
		600	8	0
		900	9	2
C	Ball Oscillating Angle (deg)	5	9	2
		10	9	0
		15	8	1
D	Cyclic Radial Pressure (psi)	0	4	0
		1000	4	0
		1500	9	0
		2000	9	3
E	Phase Angle Between C&D (deg)	0	8	2
		45	8	0
		90	10	1
F	Static Axial Load (lb)	0	9	2
		30	10	0
		60	7	1
G	Contaminants	None	6	0
		S&D	6	0
		680	5	1
		500A	6	0
		Water	3	2
-	Predicted Wear	Low	--	1
		High	--	2

TABLE H-5. SUMMARY OF TEST BALANCE
(FOR R.P. EQUATION, LUBE 25A-1-9)

CODE	VARIABLE	LEVEL	NUMBER OF SPECIMENS 35 TESTS	4 TESTS
A	Static Radial Pressure (psi)	2000C	8	1
		1000C	7	0
		0	11	1
		1000T	1	0
		2000T	8	2
B	Speed of Ball Oscillation (cpm)	300	12	2
		600	13	0
		900	10	2
C	Ball Oscillating Angle (deg)	5	11	3
		10	13	0
		15	11	1
D	Cyclic Radial Pressure (psi)	0	1	1
		1000	11	0
		1500	11	0
		2000	12	3
E	Phase Angle Between C&D (deg)	0	8	3
		45	14	0
		90	13	1
F	Static Axial Load (lb)	0	14	2
		30	10	0
		60	11	2
G	Contaminants	None	8	0
		S&D	6	0
		680	6	1
		500A	7	1
		Water	8	2

TABLE H-6. COMPARISON OF ACTUAL AND PREDICTED WEAR VALUES FOR WEAR EQUATIONS LUBE 24A-1-6 AND LUBE 24G-1-7

LOADING	BEARING DESIGNATION	TEST TIME (HOURS)	LINER WEAR (INCHES x 10^5) ACTUAL	PREDICTED	RESIDUAL	RESIDUAL/Sy[e]
Compression	DL-45 (4-3)	350	105[a]	66[c]	39	0.42
Compression	DL-49 (4-5)	250	1115[a]	796[c]	319	3.42
Compression	DL-53 (4-11)	100	450[a]	126[c]	324	3.47
Tension	DL-46 (4-4)	350	25[b]	92[d]	-67	-0.73
Tension	DL-54 (4-9)	100	1030[b]	983[d]	47	0.51
Tension	DL-53 (4-11)	100	635[b]	262[d]	373	4.05

Notes:

a. These values were obtained from Run number 4 of Table 7.

b. These values were obtained from Run number 4 of Table 8.

c. These values were predicted by equation LUBE 24A-1-6.

d. These values were predicted by equation LUBE 24G 1-7.

e. The standard error of the estimate (Sy) for equations LUBE 24A-1-6 and LUBE 24G-1-7 are 93.4 and 92.0, respectively.

TABLE H-7. COMPARISON OF ACTUAL AND PREDICTED RADIAL PLAY VALUES FOR EQUATION LUBE 25A-1-9

BEARING DESIGNATION	TEST TIME (HOURS)	RADIAL PLAY (INCHES x 10^5) ACTUAL[a]	PREDICTED[b]	RESIDUAL	RESIDUAL/Sy[c]
DL-45 (4-3)	350	65	43	22	0.26
DL-46 (4-4)	350	30	44	-14	-0.17
DL-54 (4-9)	100	640	638	2	0.02
DL-53 (4-11)	100	1085	639	446	5.32

Notes:

a. These values are a summation of liner wear on compression side from Run Number 4 of Table 7 and liner wear on tension side from Run Number 4 of Table 8.

b. These values were predicted by equation LUBE 25A-1-9.

c. The standard error of the estimate (Sy) for LUBE 25A-1-9 is 83.8